OFFICIAL PUBLICATION

JOURNAL

Heating • Refrigerating • Air Conditioning • Ventilating

AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS

JUL 14 1960

Walter A. Grant succeeded Dan D. Wile as President of ASHRAE at the 67th Annual Meeting in Vancouver. His message to members appears on page 47.

JULY 1960

NEW CONDENSER WATER REGULATING VALVE

NEVER CHATTERS

A-P MODEL 65A IS SMALL FOR EASY INSTALLATION . . .

*FEATURES SELF-CLEANING ORIFICE FOR RELIABLE OPERATION EVEN IN DIRTY WATER

The A-P model 65A eliminates chatter because it is not affected by inlet pressure variations . . . a unique design feature of the 65A valve in regulating water flow to condensing unit by head or condenser pressure.

The model 65A cleans itself as it operates. A Graphitar sliding block — the only moving part in water — wipes over the orifice in the stainless steel facing plate. Provides trouble-free service even in dirty water.

Compact size makes the A-P 65A quick and easy to install. Adjusting stem is completely accessible for easy manual adjustment — 65 to 300 psi. All moving valve parts operate as one unit. Gives quiet operation, increased bellows life. Low-friction Graphitar block won't wear out. Nylon-reinforced rubber diaphragm gives positive seal — provides flexibility for closing-opening action. Available in sizes ⅜", ½", ¾" female N.P.T.

Next time you install a condenser water regulating valve, install the industry's newest, quietest, pressure-actuated valve . . . the A-P model 65A.

Creative controls for industry

CONTROLS COMPANY OF AMERICA

HEATING AND AIR CONDITIONING DIVISION

2456 N. 32nd St., Milwaukee 10, Wisconsin • Cooksville, Ontario • Zug, Switzerland

HAC-42-60

JULY
1960

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JOURNAL

Formerly Refrigerating Engineering including Air Conditioning, and incorporating the ASHAE Journal.

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Comment

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. . A QUESTION MORE, OR A QUESTION LESS . .

Queried the late Charles F. Kettering, presumably of anybody in the services of their mutual employer who would stand still long enough to listen to such nonsense, "Why is the grass green?".

No cants or evasions like chlorophyl or refractions for Boss Kettering.

We do not recall that he ever found his answer though much research ensued.

You may recall that he also pondered how it could be that an aerodynamically impossible insect such as the bee could fly. He was a man just about bursting with such questions.

We have long felt that his genius, and such he may have come close to being, lay in seeking the real answers to apparently simple unknowns for which there existed only witch doctor answers.

Surely he did not exhaust the possibilities inherent in such an approach. We have a few of our own—

Why do stylists dominate engineers?
Must the flash boiler be forever taboo?
Why not use the most efficient refrigerant?
How relative is comfort?
Is the job ahead "total environment" meaning air conditioning plus sound plus light?

We do not pretend to know the answers to such questions but we do know that any one of them could be a key to a revolution of sorts in engineering, product design, living habits and industrial growth.

Edward R Searles
Editor

Late news highlights

BRI proceedings Now available, "New Methods of Heating Buildings" contains proceedings of that research correlation conference which was conducted as one of the programs of the BRI Fall Conferences in November 1959. Several ASHRAE members, including Vice President John Everetts, Jr., Walter F. Friend, Herbert T. Gilkey, the late Harold A. Lockhart, J. Raymond Carroll, Frank H. Faust and Richard C. Jordan participated in this conference. Copies of illustrated, 140-page Publication 760 may be obtained from the Building Research Institute, 2101 Constitution Ave., Washington 25, D. C.,; price is $5.

Questions and answers Planned to keep pace with the rapidly developing new engineering ideas and procedures represented in recent professional engineer's license examinations, *Professional Engineer's Examination Questions and Answers* by William S. LaLonde, Jr. has been published in a revised and enlarged Second Edition. Over 600 questions selected from recent actual examinations for the Professional Engineer License, the Land Surveyor License and the Engineer-in-Training Certificate are included. McGraw-Hill, 330 West 42nd Street, New York, N. Y., 615 pages, $7.50.

Oilheat industry Consisting of representatives of the National Fueloil Council and the Oil-Heat Institute of America, Inc., the Oilheat Industry Organization Committee held its first meeting in New York, N. Y., on May 12. Called for the purpose of establishing a single trade association encompassing all segments of the oil-heating industry, including refiners, manufacturers of heating equipment and distributors, the meeting made some advances towards the creation of the new organization structure, according to George Gruberg, presiding chairman.

Salary survey Fourth survey of earnings of engineering graduates is now being conducted by the Engineering Manpower Commission of Engineers Joint Council. Representing highly comprehensive data, the survey will probably cover over 200,000 engineers employed in industry, government and education. Last survey conducted in 1958 indicated that the average increase in earnings of engineering graduates was 6.5% annually from 1953 to 1958. According to the Commission, printed reports should be available at the end of the year.

Data updated At its semiannual meeting in Cleveland, Ohio recently, the Technical Data Committee of the National Warm Air Heating and Air Conditioning Association devoted a two-day session to updating the Association's Technical Data Book which serves as the source of information for material used in NWAHACA's 15-Manual Library. This session was engaged largely in preparing for presentation newly developed values of heat transfer through ducts under varying conditions.

Wind tunnel Recently completed, a multi-purpose, continuous-flow, open-circuit, atmospheric induction-type wind tunnel at Cook Electric Company is powered by two turbojet engines which generate about 5000 lb of thrust each. The two engines have provisions for extraction of compressor discharge air to test various instruments, anti-icing equipment, air conditioning and defrost systems.

Exposition committee H. T. Jarvis, president of Recold Corporation, has been named chairman of the 12th Exposition Committee of the Air Conditioning and Refrigeration Institute, it was announced by the Board of Directors of ARI. To be held February 12-15, 1962 in Los Angeles, Calif., this will mark the first time that an ARI show has been staged on the West Coast. Other members of the committee are: W. A. Siegfried, president of Superior Valve & Fittings Company, vice chairman; Sydney Anderson, Jr., manager, Los Angeles branch, Airtemp Division of Chrysler Corporation; B. E. James, president, McQuay, Inc.; J. A. Mulcahey, vice president of sales, Dunham-Bush, Inc.; George A. Schnier, national sales manager, Refrigeration Department, American Potash & Chemical Company; and George Mills, ARI show director.

UPADI in Buenos Aires

Sixth UPADI Convention will take place in Buenos Aires, Argentina, September 18-22. The program scheduled will include reports of various committees, election and installation of Convention officers, round table discussions, committee meetings, voting on resolutions and recommendations and election of Board members; additionally, numerous social events are planned.

Canadian engineers meet

74th Annual General and Professional Meeting of the Engineering Institute of Canada, held in Winnipeg, May 25-27, was highlighted by a varied and comprehensive technical program. The Institute is the largest professional society in Canada with over 20,000 members and 50 active branches extending from coast to coast.

GAMA statistics

Gas Appliance Manufacturers Association reports that January-April 1960 shipments by manufacturers of gas conversion burners, which are used in adapting existing boiler and furnace central heating systems from other fuels to gas, showed a 22.7% increase over shipments reported for the same period in 1959. There was an 11.5% increase in sales of gas-fired boilers for the first four months of this year over the same period last year. Shipments of gas-fired furnaces, including the forced warm air and gravity types, for the January-April period were decreased by 8.9%. The four-month combined totals of the three types of central heating equipment showed a 4.5% decline from last year.

25 years old

Skuttle Manufacturing Co. of Milford, Mich., manufacturer of humidifiers and filters for warm air furnaces, is celebrating its 25th anniversary. John L. Skuttle, one of the founders, is credited with designing and building the first bonnet-type humidifier known in the United States.

ASME meeting

Held in Dallas, Texas, June 5-9, the 1960 ASME Semiannual Meeting and Aviation Conference included the following papers as part of its extensive program: Toward Resolution of the Chaos and Confusion in Our System of Units by Carl F. Kayan, Columbia University; The Free Expansion of Dry and Moist Air, by James H. Potter, Stevens Institute of Technology and Martin Levy, Newark College of Engineering; Temperature Control System for Able-4 Atlas Lunar Satellite, by Roy M. Acker, R. P. Lipkis and J. E. Vehrencamp, Space Technology Laboratories, Inc.; Moisture Accumulation in Aircraft Soundproofing and Insulation, by Jorgen Jensen-Gaard, Scandinavian Airlines System; and Moisture Migration in an Aircraft Cabin Wall, by Floyd B. Bjorklund and Alvin C. Watson, Douglas Aircraft Co., Inc.

Anniversary year

To call attention to its 75th anniversary year, York Division of Borg-Warner Corporation has undertaken a nationwide "Diamond Anniversary Promotion" on York room air conditioners in all major markets.

NSF bulletin

"Science and Engineering in American Industry," containing final results of a comprehensive survey of funds and personnel for industrial research and development covering the year 1956, has been released by the National Science Foundation. NSF bulletin 59-60 may be obtained from the Superintendent of Documents, U. S. Government Printing Office, Washington 25, D. C. for 70 cents.

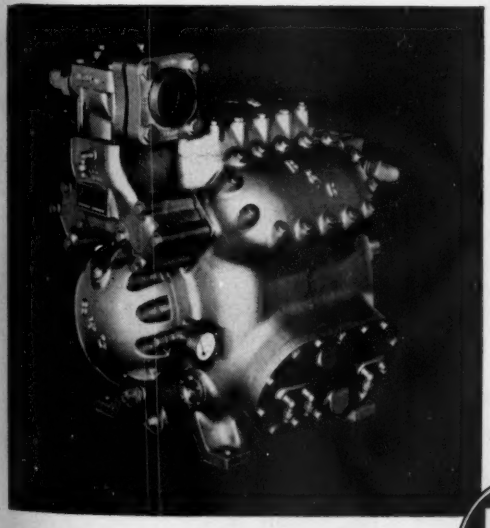
Symposium on temperature

Temperature – Its Measurement and Control in Science and Industry is the topic of a national symposium to be held in Columbus, Ohio, March 27-31, 1961, sponsored jointly by the American Institute of Physics, the Instrument Society of America and the National Bureau of Standards. Papers to be presented there will cover the meaning of the temperature concept as well as analytical theory, with appropriate emphasis on instrumentation and the engineering aspects – thermocouples and radiation pyrometry in full range of temperature. Interests from cryogenics to ultra high plasma temperature will also be included.

Temperature tolerances

Frozen ducks and geese resemble other poultry in quality retention during their commercial life, in spite of their higher fat content, it was disclosed from research conducted at the Western Utilization Research and Development Division of the U. S. Department of Agriculture's Agricultural Research Service in Albany, Calif. At 0 F commercial packs of frozen ducks and geese hold up well for 9 to 12 months, according to the researchers.

CHECKING OUT TWO 150 TON B&G® PACKAGE LIQUID COOLERS



All B&G Package Liquid Coolers and Condensing Units from 7½ to 150 tons are equipped with this new-design B&G Compressor.

Here is the final and a very important stage in the production of B&G Package Liquid Coolers. When the four men finish this check-out, the units will be shipped with *everything mechanically correct and in accordance with the requirements of the order.*

These two 150 ton Package Liquid Coolers have completely wired panels with increment starting—*virtually plug-in units!* All major parts are made in the B&G plant... resulting in a unit unmatched for efficient operation and carrying one manufacturer's guarantee!

One of their outstanding features is the compressor, a masterpiece of advanced designing and precision manufacture. It establishes an entirely new standard of performance—smoother, quieter and with minimum vibration. All compressors have *one* bore—a few interchangeable parts will service all units, reducing maintenance problems.

BELL & GOSSETT
COMPANY
Dept. GG-64, Morton Grove, Illinois

Canadian Licensee: S. A. Armstrong, Ltd., 1400 O'Connor Drive, Toronto 16, Ontario

A FULL LINE OF REFRIGERATION AND AIR CONDITIONING "PACKAGES" AND COMPONENTS

B&G Condensing Unit

B&G Evaporator

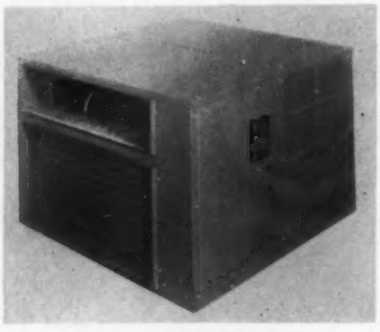
B&G Centrifugal Pump

B&G Condenser

PARTS and PRODUCTS

HEAT PUMPS

All-electric models added to this manufacturer's present line of residential and commercial self-contained and remote type air conditioning units will include such features as weather resistant finish, new design aluminum coils, central control junction box, heavy insulation, over-sized filter-drier, service valves, thermal expansion valves and high pressure control.



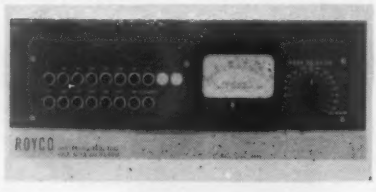
Accessory strip heater packages are being offered as optional equipment to facilitate heating loads required in areas with low winter temperatures.
Payne Company, 855 Anaheim-Puente Rd., La Puente, Calif.

UTILITY TRENCHER

Self-powered, self-propelled and portable, this unit digs two to three in. wide and a yard deep, or four in. wide a half-yard in depth. Driven by a six-speed winch mechanism, the Pup draws itself in a straight line down the path the user has set it to follow. A stake is driven at the point to where the trencher is to dig, the winch line is hooked to this stake, and the unit is turned on.
Davis Manufacturing Inc., 1500 McLean Blvd., Wichita, Kans.

TEMPERATURE MONITOR

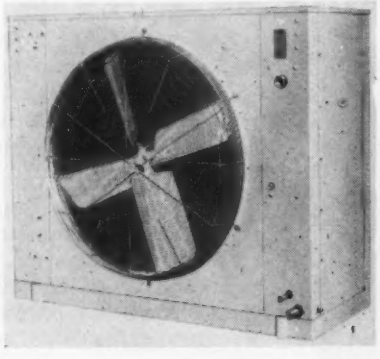
Designed to provide centralized observation of ambient temperatures in a multiplicity of areas, Type-Z Temperature-Monitor System offers a temperature range of −50 to 300 F and has no practical limitation to the number of areas which can be monitored in a single installation. Selector switches on the main panel provide for calibration and room selection. Temperatures appear directly on an indicating instrument and are cited as being accurate to ±1% of scale reading.



Sensor probes of plug-terminated types are provided and mating extension cables permit runs to be made as long as 4000 ft. Cables plug directly into the rear of the indicating panels. Central power supply of the unit operates on 115 volt, 60 cycle power and draws four watt.
Royco Instruments, Inc., 365 San Antonio Rd., Mountain View, Calif.

17-TON UNIT

Large condenser coil on this nominal 17-ton air-cooled condensing unit has ripple-edged aluminum fins flat-bonded to the copper tubes. Compressor is semi-hermetic and easily serviced in the field. It is resiliently mounted, suction cooled and overload protected.



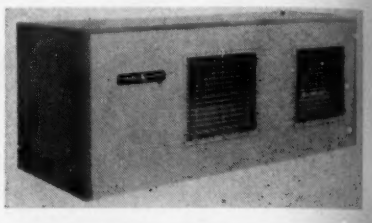
Claimed for the unit is a refrigerant circuiting method cited as improving Btu per watt performance because liquid refrigerant reaches the evaporator coil up to 20 F cooler than the condensing temperature.
Lennox Industries, Inc., Marshalltown, Iowa.

GAS HEATER

Installation and removal of this window unit is comparable to any air conditioner, since the heater does not become part of the house and may be removed whenever desired. Designated Safti-Vent Hawaiian Model, the unit operates with a sealed burner in which gas flame and pilot burn in a chamber completely sealed away from room air. A built-in vent admits oxygen from outdoors and releases products of combustion directly to the outside. Unit is approved for use with natural, manufactured, mixed or propane gases and has a 20,000 Btu capacity.
H. C. Little Burner Company, Inc., Dubois and Woodland, San Rafael, California.

SPLIT-SYSTEM UNIT

Included in this six-ton, split-system air conditioner are a weatherized outdoor remote unit, Model TA75B3 (shown), and an indoor air handling unit, Model TE72BH.



Features of the outdoor section are a high-speed compressor (3450 rpm) and belt-driven, permanently lubricated ball-bearing blowers. Fan is isolated from the compressor by a baffle, permitting ease of servicing. Blowers of the indoor section are belt-driven by a four-pole, cushion-mounted blower motor. A fan relay is furnished as standard equipment.
General Electric Company, Tyler, Texas.

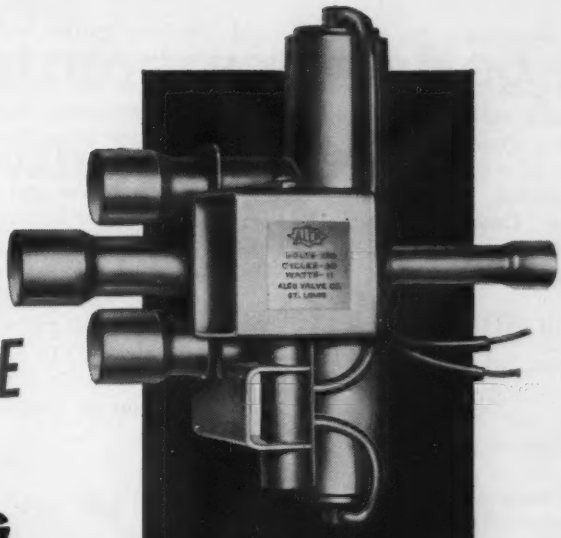
COMBUSTION ANALYZER

Analysis is provided through measurement of the carbon dioxide content of flue gas, stack temperature and draft over fire, all three measurements being made with one instrument consisting of a single meter and dial with three-scale read-out. Analyzer is available in two models, for domestic and heavy-duty industrial applications.
Victory Engineering, 603 Springfield Rd., Union, N. J.

PROTECTIVE COATINGS

Two new cold-applied protective coatings, both highly resistant to permeation by chemicals, are being produced by this manufacturer. One, Bituplastic No. 33, is a heavy duty coating suited for intermittent splash and high chemical "fall-out" areas. It can be applied by brush, roller or spray and is intended for use on structural steel, deck areas and sheet piling. Temperature tolerance is up to

Made by ALCO
OF CAST IRON AND STEEL
like a Compressor
for LONG RELIABLE LIFE



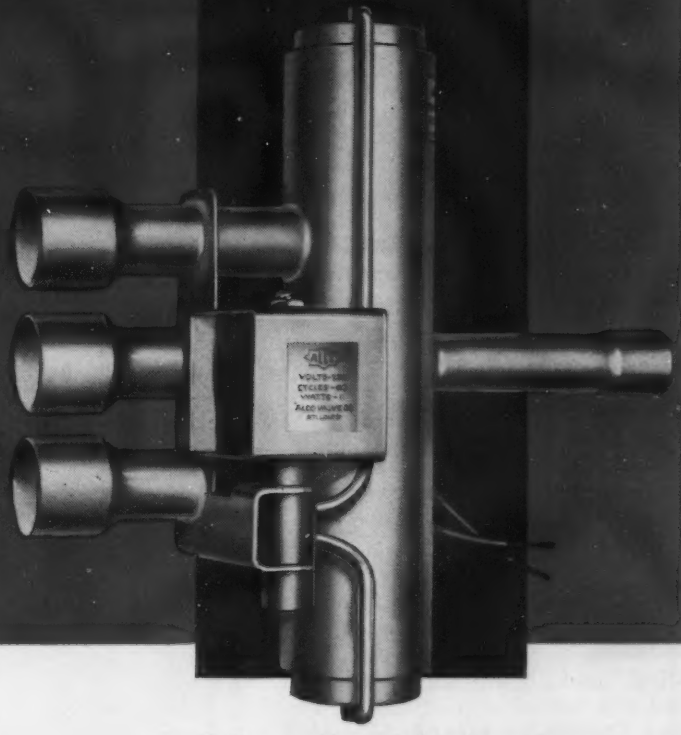
4-WAY REVERSING VALVES
for all reverse cycle systems

#4WB4-57 Capacities:
2 to 3 tons—Refrigerant 12
3 to 5 tons—Refrigerant 22

#4WB6-79 Capacities:
3 to 5 tons—Refrigerant 12
5 to 7½ tons—Refrigerant 22

Features:

- fine machining and precision honing of cast iron slide and steel outer shell—NO PLASTICS ARE USED.
- operating tests of 300,000 cycles without failure to shift and without loss of tight seating—tests made by independent laboratories.
- shifting (change cycle) while the system is in operation with a 300 PSI differential.
- rapid shifting—to shorten change-over time and therefore shorten defrost cycle.
- the shifting of the valve independent of any pressure drops, and efficient shifting under any combination of operating conditions.
- mounting in any position—except with the pilot valve upside down.



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The one complete line of refrigerant controls: Thermostatic Expansion Valves • Refrigerant Distributors • Solenoid Valves Refrigerant Filter-Driers • Suction Line Regulators • Flooded Evaporator Controls and Reversing Valves

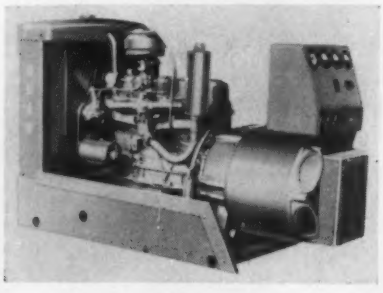
400 F. Bituplastic No. 44, the second coating, has been used to adhere cellular glass insulating blocks to storage tanks and underground piping, and as a general purpose insulation mortar. It is suitable as a top coating for insulation.

Koppers Company, Inc., Koppers Bldg., Pittsburgh 19, Pa.

GENERATOR

Now installed as standard equipment on all 25 kw gasoline-driven electric plants, the Magneciter Generator with static exciter and voltage regulator is cited as providing more efficient performance in both primary and standby applications than rotating exciters.

Advantages include: static voltage regulator, rated voltage restored within one sec, voltage fluctuation with load changes less than one-half that of rotating exciters.



D. W. Onan & Sons Inc., 2515 University Ave. S.E., Minneapolis 14, Minnesota.

BY-PASS VALVE

Eliminating intricate piping connections for induction and fan coil air conditioning systems in which water is circulated, this single control by-pass valve combines the functions of inlet and outlet stop valves and the by-pass or flushing valve which normally are installed at each coil in a central air conditioning system. One 90-deg turn of the control handle opens the by-pass and shuts off the water supply to the coil and thermostatic control. Measuring 2-1/16 x 2-5/16 in., it is available in sizes to accommodate ½ or ⅝-in. OD copper tubing with either soldered or flare type connections.

Clayton Mark & Company, 1900 Dempster St., Evanston, Ill.

REFRIGERATION GAUGE

Serving as a basic test instrument for refrigerator service work, this refrigeration gauge has external zero set and has been designed with rugged construction to withstand occasional over-pressures, sharp blows and falls.

Gauge is available in two models: 1780, which has a phosphor bronze bourdon tube and 1785, with a beryllium-copper bourdon tube. Range of Model 1780 is zero to 400 psi with scale graduations in units of ten lb, that of Model 1785 is 30 in. to zero to 60 to 250 psi (overpressure protection from 60 to 250 psi).

United States Gauge Div, American Machine & Metals, Inc., Sellersville, Penna.

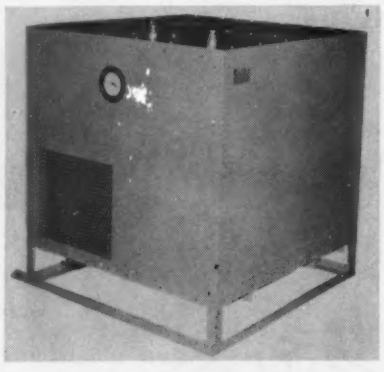
AIR CONDITIONERS

Capacities of 36,000 and 47,000 Btu/hr are cited for Models AP3-36 and AP3-47, recently added to this line of self-contained air conditioners. Both units are air-cooled and designed for operation at ambient temperatures in excess of 125 F. Low silhouette for installation in gable, crawl space or attic locations is featured. Indoor air section is completely insulated and the entire unit is designed for outdoor installation. Outdoor condenser air is admitted through louvers on both sides of the unit and discharged through the condenser to the rear by an axial fan.

Holly-General Div, Siegler Corporation, Los Angeles, Calif.

AIR DRYER

Basic components of Model DHD-20 are two cast aluminum heat exchangers, drip leg and centrifugal stripper. Operation is automatically to lower dew point of a compressed air line to approximately zero F with min line pressure drop. Unit is designated for use at ambient temperatures of 40 to 140 F.

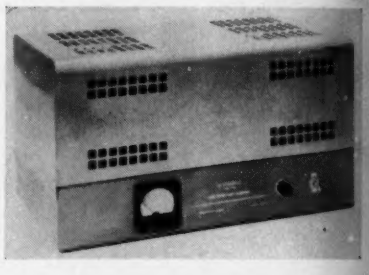


Warm line air from the air compressor enters the first heat exchanger and flows through copper tubing to a drip leg positioned at the bottom outlet of the exchanger. Air passing to the second heat exchanger is chilled and then passes through a centrifugal stripper at 40 F. Chilled air is returned to the first heat exchanger, entering at the bottom and counter-flowing to the top exit. Incoming air is reduced at this point from 170 to 110 F, increasing leaving dry air temperature from 40 to 140 F. Following each step in cooling, moisture is extracted through a balanced trap valve.

Drayer-Hanson Div, National-U. S. Radiator Corporation, 3301 Medford St., Los Angeles 63, Calif.

STATIC GENERATOR

Encased in a ventilated steel housing, this compact static generator, made especially for use in the Statronic system of dirt and dust control, provides high voltage static electricity for a max of 10,000 cfm air flow. Operating conditions at the screens through which air flows in an air conditioning system are indicated on a static voltage meter.



Electrical input required is 115 volt, 60 cycle, single-phase from a source providing 75 watt. Input source is interlocked through the main power supply to the air conditioner fan so that power is furnished to the generator only when the fan is operating.

CRS Industries, Inc., 1405 Locust St., Philadelphia 2, Pa.

PUMP-MOTOR COMBINATION

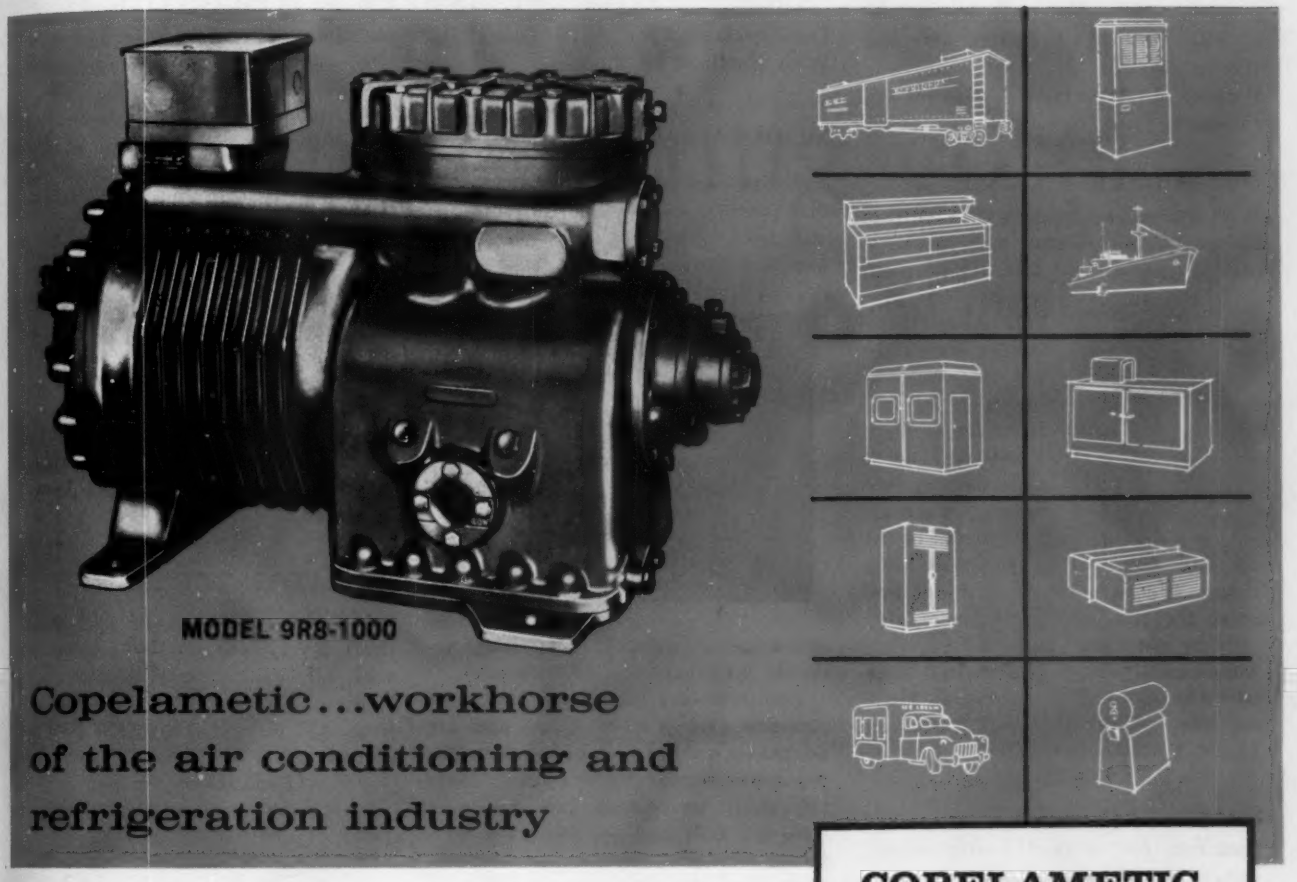
Extending this company's coverage of compact pump and motor combinations for use in oil burning, fuel transfer and other services is Series 56, which includes motors in the Nema 56 series ranging from ¼ to one hp in either single or three-phase. These may be combined with a selection of five different pumps with capacities from 20 to 360 gph.

Tuthill Pump Company, 939 E. 95th St., Chicago, Ill.

DIAL THERMOMETERS

Mercury-actuated dial thermometers for duct applications feature ease of reading and accuracy throughout the operating range, a uniformly gradu-

To keep your customers satisfied Copeland has what it takes!



Satisfied customers are our most valuable asset and convincing proof that Copeland has what it takes.

Over the years, manufacturers and dealers have ordered and re-ordered the Copelametic Accessible Hermetic, mainstay of the Copeland line. They have found Copelametics provide the outstanding performance features and lasting dependability essential to the success of a product or an installation.

This enthusiastic acceptance has enabled us to produce Copelametics in growing numbers for a wider variety of applications.

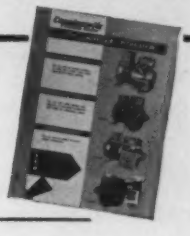
Today's Copelametic is basically the same, high quality motor-compressor introduced 21 years ago . . . relatively unchanged because the original design continues to meet the most exacting requirements. Years of service in the field have proved its suitability for all applications . . . high, commercial, low and extra-low.

Investigate the advantages of Copelametics for your products or installations.

COPELAMETIC FEATURES

- Complete accessibility to all working parts permitting on-the-job service and parts replacement.
- Positive lubrication systems.
- High output per watt-hour.
- No belts, seals or manual oiling, thus eliminating causes of 90% of motor-compressor failures.
- Statically and dynamically balanced rotating parts.

AVAILABLE in air-cooled, water-cooled, and suction-cooled models. Including new four-cylinder and six-cylinder models, sizes range from ¼ H. P. through 30 H. P. Request Specification Sheet 5907.



SINCE 1918 **Copeland** REFRIGERATION CORPORATION, Sidney, Ohio

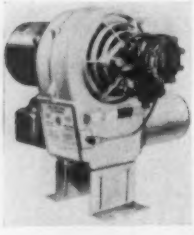
ated scale, micrometer adjustable pointer, all-stainless steel system and a ten to twelve-in. sensitive stainless steel bulb with union connection flange.

Case design permits installation for adjustable angle, rigid or remote reading. Standard ranges are: −40 to 110, −20 to 110, zero to 100, 30 to 150, 30 to 240, 100 to 400 and 100 to 800 F.

Weksler Instruments Corporation, Freeport, N. Y.

OIL BURNER

With design and construction features for simple installation and for wide use in conversion and replacement projects, Model DH Arcoflame burner meets firing rate requirements from 0.65 to three gph. Featured is a slide damper permitting accurate adjustment of the oil and air mixture without tools. Four disc-turbulator combinations cover the complete firing range, simplifying selection.



American Radiator & Standard Sanitary Corporation, Plumbing & Heating Div, 40 W. 40th St., New York 18, N. Y.

SHADE CLOTH

Specified for vertical louver blinds, this dimensionally stable type of opaque window shade cloth is made of balanced, heavy vinyl coatings on both sides of a glass cloth base. In dead white for high solar heat reflectivity, Tontine Triglas cloth is cited as testing approximately three times stronger in breaking strength than conventional, cotton-based shade cloth.

E. I. du Pont de Nemours & Company, Wilmington, Del.

LOW SILHOUETTE UNIT

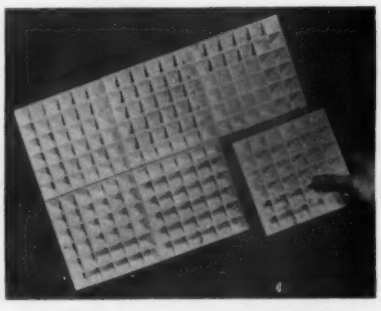
Available in five basic sizes, ten through thirty ton, the LSCU con-

densing unit has been specifically designed for air conditioning applications requiring the use of water savers. Basically, the unit is a standard LSBC blower condenser, complete with a Brunner-Metic motor-compressor, Heat-X liquid receiver, electrical controls and refrigeration accessories mounted in a specially designed section for field attachment or remote installation.

Dunham-Bush, Inc., 179 South St., West Hartford 10, Conn.

MODULAR DIFFUSER

Combinations of three types of modules provide a number of air patterns and sizes. Modules are of high impact and flame retardant plastics, with



dimensional stability, and snap securely into heavy-gauge frames.

Diffusers are designed for most types of ceiling construction and are adaptable to tile and metal pan suspension systems. Basic diffuser faces are sized to replace one or more standard tiles and are provided with frame modifications and panel assemblies dimensioned for exposed suspension.

Carnes Corporation, Verona, Wisc.

COPPER TUBE

For refrigeration and air conditioning applications, ACR (Type L) is a cleaned and sealed tube available in 20-ft straight lengths in sizes ranging from ¼ to four in. nominal OD.

Three new sizes of tube have been added to the coiled refrigeration service tube line, allowing a greater range for the user. Included are sizes with ODs of ⅞, 1⅛ and 1⅜ in. Like the 20-ft straight lengths, these 50-ft coils are cleaned and capped.

Wolverine Tube Div, Calumet & Hecla, Inc., 17200 Southfield Rd., Allen Park, Mich.

PUMP

Using a positive displacement pumping action to "squeeze" water upward, the Hi-Lift pump has a corkscrew-like rotor which rotates within a contoured stator. Action of the rotor creates a positive displacement action, moving a constant flow of water to the surface.

Unit can be operated at one-half the normal pump speed (1760 rpm) for greater operating and maintenance economy and longer pump life. Further, sandy particles which may become momentarily depressed in the rubber when the rotor passes over it are removed by the water flow as the rotor continues to turn, commending this pump for sandy wells. Rotor is of stainless steel and is protected from wear by a heavy plating of hard chrome and shaft is equipped with non-corrosive, chrome-plated bearing sleeves.

Four series are currently available, with capacities ranging from 500 to 3300 gph with lifts of up to 1000 ft and pressures to 50 psi. Pumps will fit in wells with four-in. inside diam and larger. All types of drives may be used.

Peerless Pump, Hydrodynamics Div, Food Machinery and Chemical Corporation, 301 W. Ave. 26, Los Angeles 31, Calif.

VERSATILE BOILER

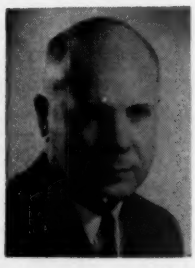
Adapted for both forced and natural draft, XWL Series Boilers can be fired with oil, gas or combination oil-gas burners. Eleven sizes of forced draft from 427,000 to 1,928,000 Btu and eleven sizes of natural draft from

297,000 to 1,350,000 Btu comprise the series. Construction is gas-tight throughout, which includes the gasketed front flue and combustion chamber doors. An insulated flush jacket is provided for min heat loss. Boilers operate on low pressure 15 psi steam and 30 psi water and are designed for instantaneous tankless coils for heating domestic hot water the year round.

Portmar Boiler Company, Inc., 193 Seventh St., Brooklyn 15, N. Y.

JOURNAL
OFFICIAL PUBLICATION
JULY 1960

A comparative study of the Manufacturing costs of thermoelectric and mechanical refrigerating systems



D. W. SCOFIELD

P. F. TAYLOR

L. A. STAEBLER
Member ASHRAE

The thermoelectric effects discovered by Seebeck, Peltier and Thomson during the 19th century are well known to engineers and scientists. However, until approximately five years ago the practical application of these effects was restricted almost exclusively to the use of thermocouples for temperature measurement. Over the years a small devoted band of scientists, Telkes, Justi, Wall, Altenkirch, Euchen, Forsborg, to name a few, applied themselves to investigations of these phenomena in the hope of discovering new materials in which the thermoelectric effects were of sufficient magnitude to extend their application beyond that of the measurement of temperature. This eventually led to the consideration of semiconductors, culminating in the work of A. F. Ioffe.[1]

During the past five years progress in this field has accelerated. The discovery and development of the transistor and other solid state devices has stimulated the need for a deeper understanding of the electrical and thermal processes in semiconductors. This has led to the detailed study of Bi_2Te_3, a material first suggested by Goldsmid[2] for thermoelectric applications. The study of this material[3,4,5,6] and associated alloys[7,8,9] has led to the development of practical devices for thermoelectric cooling. The present state of the art has received wide attention[10,11,12,13].

Thermoelectricity as applied to heat pumping has certain advantages over the mechanical system. These advantages include, (a), the elimination of moving parts except perhaps for a small fan to assist the heat transfer processes; (b), a reduction in noise; (c), the substitution of simple electrical wiring for the hermetically sealed system required to contain the refrigerant;

Increased use of thermoelectricity may in the future have applications for devices of capacities up to 200 Btu/hr and within just a few years make available thermoelectric refrigerators of up to 4 cu ft despite competition of the low cost mechanical refrigerating system.

The authors offer a comparison of the manufacturing costs of these two systems which combines improved thermoelectric materials, simplified manufacturing techniques and low cost power supplies. Here is how TE refrigeration could be economically competitive with mechanical refrigeration for several major large appliances.

D. W. Scofield is Project Engineer, P. F. Taylor is Project Scientist, and L. A. Staebler is Manager, Advanced Development Department of the Appliance Div, Philco Corp. This paper, here condensed only as to the omission of certain detailed costs and related equations, was presented at the 67th ASHRAE Annual Meeting, Vancouver, B. C., June 13-15, 1960.

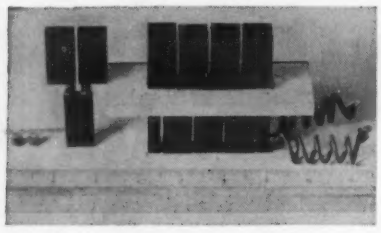


Fig. 1 Section of thermoelectric cooling unit and its components

(d), modulation of heat pumping capacity accomplished readily through a corresponding modulation of the operating current; (e), reverse cycling easily achieved by current reversal; and (f), the ease with which the thermoelectric system can be used for small capacity cooling applications.

Despite these advantages it is obvious that thermoelectric heat pumping must be competitive in manufacturing cost in order to replace the conventional mechanical system to any significant degree in the Consumer Appliance Market.

The purpose of this study was to compare the manufacturing costs of thermoelectric heat pumps with those of conventional mechanical systems for various design capacities. In order to achieve this we have developed a relationship between cost and heat pumping capacity for a thermoelectric system. The cost of the basic semiconductor materials used in establishing this relationship has been that which present manufacturers of these materials consider to be attainable within the next few years. The data obtained from this study were then used to relate manufacturing cost to system performance.

The cost figures for mechanical refrigeration systems are well established and will probably remain relatively stable during the next few years. Comparable information concerning costs and future trends for the thermoelectric system is not generally available. This study is an attempt to develop such data.

PRELIMINARY DESIGN FOR COSTING

In order to derive detailed thermoelectric costs a working system first had to be designed and built. An illustration of the make-up of a section of the cooling unit is shown in Fig. 1. The system selected was designed for 100 Btu per hr for hot and cold junction operating temperatures of 90 and 40 F, respectively. The design equations[1] used were:

$$Qi = I\,T_c\alpha - \tfrac{1}{2}\,I^2R - C\,\Delta T$$
$$W = I^2R + I\,\alpha\,\Delta T$$
$$\phi = Qi/W$$
$$N = Q/Qi$$

The symbols are defined in the Nomenclature.

Preliminary design of the TE couple geometry and selection of operating current quickly revealed that these design parameters could have a profound effect on the total manufacturing cost of the refrigerating system. To illustrate this point the cost data derived later in this study have been used to plot the curves shown in Fig. 2.

A minimum manufacturing cost can be obtained by designing the system to operate at some appropriate condition between that of maximum C.O.P. and maximum heat pumping capacity. Because the C.O.P. may also be an important consideration a compromise can then be made between minimum manufacturing cost and maximum C.O.P. This matter is covered in detail later, and is illustrated in Fig. 3.

The unit under consideration was designed with this compromise in mind and subsequent calculations based upon the cost data provided from the above approach showed that the selected operating conditions and TE couple geometry were close to optimum.

The preliminary design calculations assumed, (a) TE couple legs ¼ in. long by ¼ in. diam, (b) negligible contact resistance between the semiconducting material and the copper connecting strips in the couples, and (c) a TE material with a Figure of Merit (Z) of $3 \times 10^{-3}\ C^{-1}$. The various parameters contributing to the Z of this material are shown below. They are representative of the best TE materials presently available.

Seebeck coefficient = ± 200 μv/C
Electrical resistivity = $0.83 \times 10^{-4}\ \Omega$ cm
Thermal conductivity = 1.6×10^{-3} W Cm^{-1} C^{-1}

TABLE I

PRELIMINARY DESIGN INFORMATION

Design Capacity—Btu per/hr	100	300	1000
Number of TE Couples	25	66	215
Current (Amp), dc	21.4	25.6	27.2
Voltage (Volt)	2.1	6.4	22.0
Power Input (Watts)	44	163	598
C.O.P.	0.67	0.54	0.49
Heat to be Dissipated—Btu per/hr	250	857	3040

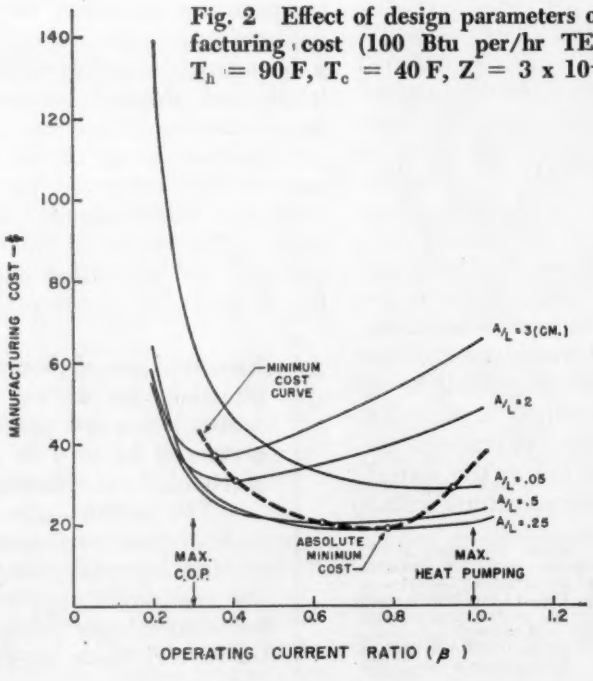


Fig. 2 Effect of design parameters on manufacturing cost (100 Btu per/hr TE system. $T_h = 90$ F, $T_c = 40$ F, $Z = 3 \times 10^{-3}\ C^{-1}$)

TABLE II

PRELIMINARY COST INFORMATION

Design Capacity—Btu per/hr ..	100	300	1000
Power Supply Costs	$ 8.67	$13.29	$ 26.77
TE Material Costs	7.50	19.77	64.60
Other Material and Labor Costs	1.75	4.63	15.00
Fixed Costs	3.00	5.00	7.00
Total Costs	$20.92	$42.69	$113.37

On the basis of the foregoing, the number of couples required and the characteristics of the power supply were determined. In addition, the heat to be dissipated from the hot junctions was calculated and from this information an actual system was designed consisting of an array of thermoelectric couples with heat exchange fins on both the hot and cold junctions.

An air moving system was incorporated which employed a small motor with double ended shaft and two fans, one to circulate the air over the cold fins in the refrigerated enclosure and the other to pass ambient air over the hot fins. The same design considerations were then extended to 300 and 1000 Btu per hr capacity systems.

Representative design information developed for costing purposes is shown in Table I.

PRELIMINARY COSTS

From design information given in Table I the Philco Cost Engineering Department developed reliable manufacturing cost data for both the power supply and the refrigerating unit for design capacities of 100, 300 and 1000 Btu per hr. The entire procedure for all three capacities was repeated assuming thermoelectric material with a Figure of Merit of 2 x 10^{-3} C^{-1}. This was done because at the time of the study such material was then obtainable from several sources in quantities sufficient for full scale device construction.

In order to predict future costs based on anticipated improvements in thermoelectric materials, the costing procedure was also repeated with Figures of Merit of 5 x 10^{-3} C^{-1} and 10 x 10^{-3} C^{-1}, with each of the basic parameters improved in approximately equal proportions. This appeared to be the best procedure since it is difficult to predict in which of the parameters the major improvement may come.

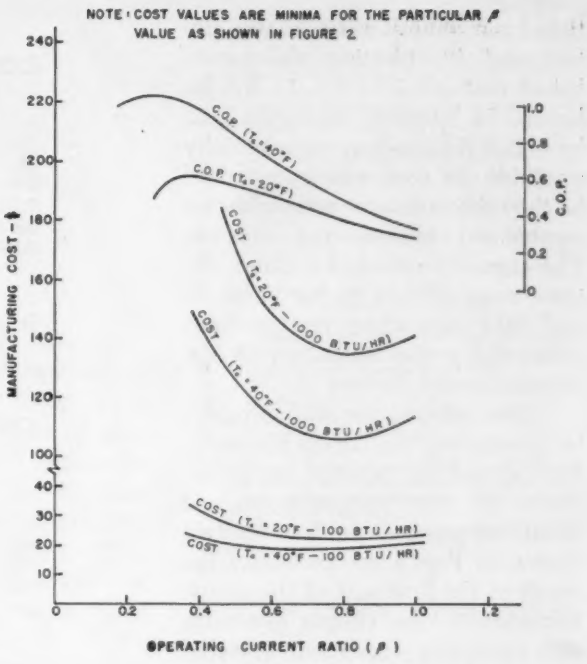

NOTE: COST VALUES ARE MINIMA FOR THE PARTICULAR β VALUE AS SHOWN IN FIGURE 2.
MANUFACTURING COST — $
C.O.P.
C.O.P. (T_c = 40°F)
C.O.P. (T_c = 20°F)
COST (T_c = 20°F — 1000 BTU/HR)
COST (T_c = 40°F — 1000 BTU/HR)
COST (T_c = 20°F — 100 BTU/HR)
COST (T_c = 40°F — 100 BTU/HR)
OPERATING CURRENT RATIO (β)

Fig. 3. Effect of the operating current ratio (β) on manufacturing cost and C.O.P. for 100 and 1000 Btu per/hr TE systems. ($T_h = 90$ F, $T_c = 40$ F, and 20 F)

The cost information developed for the refrigerating system using a TE material with a Figure of Merit of 3 x 10^{-3} C^{-1} is shown in Table II.

From Table II it was possible to write the following equation for the cost of the power supply:

$$\text{(Power Supply Cost)} = \$2.00 + \$0.25I + \$0.03W$$

This includes a term for the fixed cost, a second term related to operating current (I) and a third term relating to watt (W).

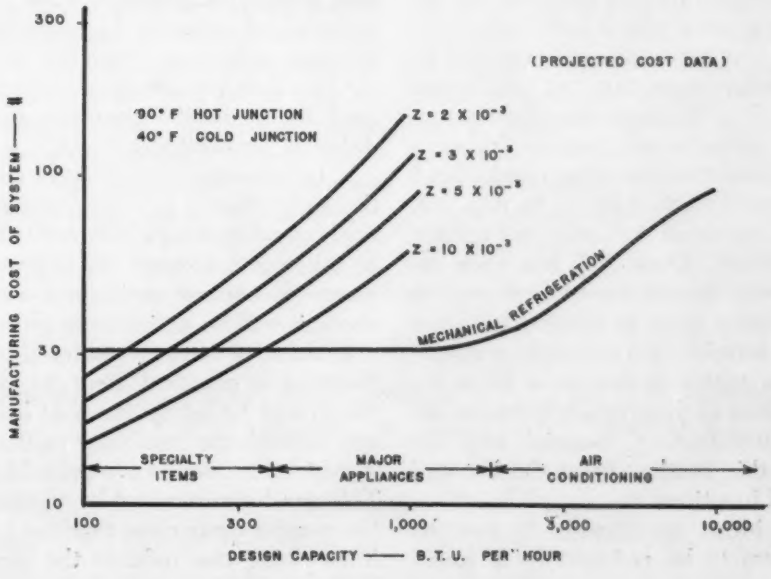
A similar equation was written for the cost of the refrigerating unit itself:

$$\text{(TE Refrigerating Unit Cost)} = \$(\text{fan}) + \$0.3N + \$0.07N$$

This includes a fixed term for the fan ($3.00 for 100 Btu per hr, $5.00 for 300 Btu per hr, and $7.00 for the 1,000 Btu per hr system), a second term for the TE materials involved and a third term for the other materials and assembly. Both the second and third terms are related to the number (N) of TE couples involved.

The cost curve for conven-

Fig. 4 Comparative manufacturing costs of thermoelectric vs mechanical refrigeration systems


(PROJECTED COST DATA)
90° F HOT JUNCTION
40° F COLD JUNCTION
Z = 2 X 10⁻³
Z = 3 X 10⁻³
Z = 5 X 10⁻³
Z = 10 X 10⁻³
MECHANICAL REFRIGERATION
MANUFACTURING COST OF SYSTEM — $
SPECIALTY ITEMS
MAJOR APPLIANCES
AIR CONDITIONING
DESIGN CAPACITY — B. T. U. PER HOUR

tional mechanical systems was determined by plotting the established costs of 1/12 hp, ⅛ hp, ¼ hp and ⅜ hp units. Since the 1/12 hp unit is the smallest commercially available its cost was assumed to be the minimum now attainable for mechanical refrigerating systems. The capacity ratings for these systems were related to the same 40 and 90 F operating temperatures selected for the operation of the thermoelectric devices.

This information was sufficient for producing the curves shown in Fig. 4 and to proceed with the study of cost minimization. It should be pointed out that the data shown in Fig. 4 are primarily the result of the first part of the study. Fortunately, the couple geometry and operating conditions selected for calculating this data were intuitively so well chosen that the use of the data deduced from the cost minimization procedures subsequently described do not change the curves of Fig. 4 significantly.

COST MINIMIZATION PROCEDURE

When considering the design of a thermoelectric heat pump of any specific capacity, once the couple leg length L has been decided upon, there are two basic variables which ultimately determine the cost. As indicated previously, these are current (I) and the ratio of the cross-sectional area to length of the couple legs (A/L). There are an infinite number of combinations of these two variables which yield the same heat pumping capacity.

There are two particular modes of operation which have been thoroughly treated mathematically, namely, operation at maximum C.O.P. and at maximum heat pumping capacity per couple. For each of these two operating conditions I is specified in relation to A/L but the choice of A/L can still remain arbitrary. Once A/L has been selected, the choice of L is not so arbitrary since its value determines the deterioration in couple performance which is due to a finite resistance at the contacts between the semiconducting material and the metallic conductors at the hot and cold junctions.

More specifically, Z can be shown to be reduced by a factor $\left(1 + \frac{2\rho_c}{\rho L}\right)$ due to this contact resistance in a couple, where ρ and ρ_c are respectively the semiconductor resistivity and the contact resistance for 1 cm². Thus the designer must resign himself to a certain deterioration in Z based on the known values of ρ and ρ_c and then select L accordingly. The importance of reducing ρ_c is readily apparent when one bears in mind the fact that the volume of material used for a specific operating condition is proportional to L^2.

In selecting couple geometry there are other important and practical considerations which must also be taken into account. As improvements in contact resistance come about it will be desirable to reduce L in order to conserve material but there is a practical limit to this which will be set by the heat leakage through the insulating medium in which the couples are imbedded. This can be minimized by packing the couples quite close together but in so doing one reduces the basic area available for heat transfer on the two sides and this may present an additional practical problem.

It is readily apparent that as I and A/L are increased, the number of couples (N) required for a given total capacity unit decreases, and vice versa. It should be therefore possible to determine a combination of I, A/L and N which yields a minimum cost figure.

This section covers such a procedure and also shows the relationship between cost and C.O.P. This is valuable information since both are important factors to be considered in the final design.

TABLE III

COST OF 100 BTU PER/HR SYSTEM AS A FUNCTION OF β (T_H = 90 F, T_c = 20 F)

β	Power Supply Cost	Fixed Cost	Assembly Cost	T.E. Material Cost	C.O.P.	A/L	Total Cost
1.0	$9.08	$3.00	$3.91	$7.49	0.28	0.244	$23.48
0.8	7.86	3.00	3.62	8.00	0.39	0.282	22.48
0.6	7.18	3.00	3.51	10.00	0.53	0.365	23.69
0.4	7.21	3.00	3.78	17.52	0.62	0.587	31.51

It thus appears that the absolute minimum cost is about $22.48 with β = 0.8 and A/L = 0.282.

TABLE IV

COST OF 100 BTU PER/HR SYSTEM AS A FUNCTION OF β (T_H = 90 F, T_c = 40 F)

β	Power Supply Cost	Fixed Cost	Assembly Cost	T.E. Material Cost	C.O.P.	A/L	Total Cost
1.0	$8.03	$3.00	$3.49	$5.72	0.34	0.209	$20.24
0.8	6.96	3.00	3.21	6.01	0.50	0.240	19.18
0.6	6.28	3.00	3.02	7.14	0.71	0.301	19.57
0.4	5.92	3.00	2.76	8.91	.99	0.412	22.80

In this case the minimum cost is about $19.18 with β = 0.8 and A/L = 0.240.

TABLE V

COST OF A 1000 BTU PER/HR SYSTEM AS A FUNCTION OF β (T_H = 90 F, T_c = 20 F)

β	Power Supply Cost	Fixed Cost	Assembly Cost	T.E. Material Cost	C.O.P.	A/L	Total Cost
1.0	$46.15	$7.00	$12.35	$74.90	0.28	0.772	$140.40
0.8	35.92	7.00	11.42	80.00	0.39	0.893	134.34
0.6	29.95	7.00	11.15	100.00	0.53	1.157	148.10
0.4	28.28	7.00	11.98	172.52	0.62	1.86	222.78

CONCLUSIONS

The purpose of this study was to compare the manufacturing costs of thermoelectric and mechanical refrigerating systems for various design capacities. These comparative costs are shown in Fig. 4. In order to permit this comparison a relationship between cost and heat pumping capacity was developed for the thermoelectric system. This relationship was developed by writ-

TABLE VI

COST OF A 1000 BTU PER/HR SYSTEM AS A FUNCTION OF β (T_H = 90 F, T_c = 40 F)

β	Power Supply Cost	Fixed Cost	Assembly Cost	T.E. Material Cost	C.O.P.	A/L	Total Cost
1.0	$38.47	$7.00	$11.07	$52.70	.34	.661	$113.74
0.8	29.67	7.00	10.17	60.10	.50	.760	106.94
0.6	23.96	7.00	9.66	72.40	.71	.953	113.02
0.4	20.43	7.00	9.63	108.40	.99	1.304	145.46

ing a cost equation for the complete TE system, including power supply, thermoelectric couples and heat exchange surfaces. The variables involved include total Btu per hr design capacity, TE material parameters, operating current and the geometry of the thermoelectric couple. From this equation a relationship between couple geometry and the operating current for minimum cost was obtained. Then, by selection of various operating currents an absolute minimum cost was deduced. This allowed us to relate manufacturing cost with C.O.P. and arrive at a design compromise between absolute minimum cost and maximum C.O.P.

The cost of the TE materials used in establishing the relationship is somewhat below the present market price. However, it is one which companies presently engaged in the fabrication of such materials consider to be attainable within the next few years. At present, materials with $Z = 3 \times 10^{-3}\ C^{-1}$ are commercially available at approximately $0.22 g in production quantities, as compared with the $0.085 g assumed in these calculations.

Since the high purity elements used in the fabrication of Bismuth ternary material cost only about $0.05 g before processing, a final cost somewhat less than $0.10 g may well be realized in the near future. Other costs are based upon present day technology of TE couple fabrication and upon conventional design of the dc supply.

If significant improvements are made in the Figure of Merit of the TE materials or in couple fabrication, either of which will permit the use of less TE material for the same performance, then thermoelectricity will become more competitive than this study indicates. Further cost reduction may likewise be possible in dc power supplies.

Already the use of thermoelectricity for military and other special applications is increasing rapidly. This is evidenced by the government support enjoyed by many companies working in this field. In the consumer appliance industry economic factors become of prime importance. However, despite the competition of the low cost mechanical refrigerating system, thermoelectricity may have applications in the near future for devices of capacities up to 200 Btu per hr, and within just a few years thermoelectric refrigerators of up to 4 cu ft may be available to the public.

Although it does not appear that thermoelectricity will in the foreseeable future completely replace or obsolete the mechanical system, there are good reasons to expect that continued research will eventually provide a combination of improved thermoelectric materials, simplified manufacturing techniques, and low cost power supplies which will make thermoelectric refrigeration economically competitive with mechanical refrigeration for the larger major appliances.

NOMENCLATURE

α = Net Seebeck Coefficient, V/C
ρ = Resistivity (Ω cm)
K = Thermal Conductivity (W Cm^{-1} C^{-1})
A = Couple leg cross-section area (cm^2)
L = Couple leg length (cm)
G = A/L (cm)
Φ = Coefficient of performance C.O.P.
Qi = Heat pumping capacity of a couple (Watts)
Q = Heat pumping capacity of a unit (Watts)
W = Total power to operate a unit (Watts)
N = Number of couples in a unit
I = Current (amps)
β = Operating current ratio = Operating current / Current for maximum heat pumping
T_H = Hot junction temperature (K)
T_c = Cold junction temperature (K)
ΔT = Th — Tc (K)
R = Couple resistance (ohms)
C = Couple thermal conductance (WC^{-1})
F = Fixed cost of a unit ($)
P = Power supply cost = f + aI + bW where f is a fixed cost and a and b are costs related to the current and power requirements of the supply
E = Assembly cost per couple
M = Semiconductor material cost per cm^3
S = Total cost
Z = Figure of merit

THEORY AND CALCULATIONS

The numerical values of the cost constants were taken as follows (see Table II):

F = $3.00 for 100 Btu/hr unit
= $5.00 for 300 Btu/hr unit
= $7.00 for 1,000 Btu/hr unit
f = $2.00
a = $0.25
b = $0.03
E = $0.07
M = $0.68

Now S = Fixed cost + power supply cost + assembly cost + material cost

$$= F + f + aI + bW + NE + 2NALM \quad (1)$$

W and N are readily expressed in terms of I. By introducing the operating current ratio β, such that $I = \beta T_c \alpha / R$ the cost equation can be written in terms of the variables β and G. Differentiating the equation in this form with respect to G and equating the result to zero gives the value of G yielding minimum cost for a particular mode of operation, i.e., a given β. Variation of β then leads to a value of the absolute minimum cost. This procedure leads to the following expressions for each of the terms in the cost equation.

(*Continued on page 72*)

ASHRAE NATIONAL MEETINGS AHEAD

1961	1962	1963
Feb. 13-16 Semiannual Chicago, Ill.	Jan. 28-Feb. 1 Semiannual St. Louis, Mo.	Feb. 11-14 Semiannual New York, N. Y.
June 26-28 Annual Denver, Colo.	June 25-27 Annual Miami, Fla.	

The dual solution system

—a different approach to air conditioning

F. H. HIBBERD

The dual system

. . . . provides independent control within its capacity . . . .

. . . . avoids the difficulties of wet cooling coils after compressor shut off on intermittent operation . . . .

. . . . does not require a large increase in electrical service or consumption . . . .

. . . . utilizes the same equipment for heating and cooling . . . .

. . . . avoids heating and evaporation from an aerated and contaminated solution . . . .

. . . . requires a far smaller cooling tower or its equivalent.

. . . . there are no pressures at any time, there is no hazard from rupture

Essentially, the dual solution system is of the open absorption type. Air to be conditioned is brought into direct contact with a hygroscopic solution. The dual solution system differs from other open absorption systems in that the hygroscopic solution is cooled continuously so that it simultaneously cools and dehumidifies the air. As both strength and temperature of the solution are controllable readily within limits, the temperature and moisture content of the conditioned air likewise are controllable individually. This patently valuable asset is secured, however, at the cost of a relatively complex cycle and apparatus.

In discussing various types of absorption refrigeration methods, differing meanings have been assigned to the terms used to denote the condition of the solutions at various points in the cycle. Here the term "strong" is taken to mean a solution which is relatively concentrated as to its hygroscopic component. For example, a solution containing 40% lithium chloride is said to be relatively strong while one containing 15% is said to be relatively weak.

The system described herein is not limited inherently to a specific hygroscopic material. Furthermore, as there are in fact two separate solution circuits it is possible, probably for some applications desirable, to use different substances in the two circuits. However, for reasons hereinafter discussed, lithium chloride has been used in the experimental equipment and will be assumed in discussing the operation.

In Fig. 1, the hygroscopic, relatively weak, primary solution cools and dehumidifies the air being conditioned in the contactor. Having absorbed sensible heat and been diluted it is drawn by vacuum into the vapor exchanger where it loses water by evaporation resulting in cooling and restoration of strength. It is removed continuously by a pump and reused.

Conditions in the vapor exchanger favor evaporation from the primary solution because a vacuum is maintained therein and a fan continuously recirculates air between contact with the sprayed primary solution and a spray of secondary solution, which although relatively warm is relatively strong enough to absorb the water vapor picked up from the primary solution. Thus, the primary solution is continuously cooled and reconcentrated while the secondary solution is continuously warmed and diluted. The degree of vacuum is not critical for this purpose but it must be high enough to minimize the carry back of sensible heat by the air stream from the warm secondary to the cold primary solution. In practice a vacuum of about 27.5 in. is satisfactory.

The secondary solution is continuously recooled by passing it through the solution cooler where it is in indirect heat exchange with a stream of water saturated outdoor air. It is reconcentrated by withdrawing a small portion—about 6% of total flow—and passing through a double effect evaporator. In the experimental unit the first effect was oil fired. Gas fuel would be suitable and more easily controlled. As the first evaporator is maintained at a temperature approaching 260 F, steam could only be used at a pressure of 25 psig or more.

Fig. 1 shows diagrammatically the arrangement of the double effect evaporator system. Feed to the first evaporator is preheated by two countercurrent heat exchangers in series. The first evaporator is maintained at about 10 in. vacuum by a small combined condensate and vacuum pump. The second effect evaporator is at approximately vapor exchanger vacuum which is maintained by a sliding vane pump controlled by a vacuum switch.

The heating and humidifying cycle, although it utilizes much of the same equipment, is somewhat more orthodox. By shifting valves and dampers the solution cooler becomes the means of heating air to be conditioned. The solution (actually that which is the "secondary"

F. H. Hibberd is with the Air Conditioning Development Company. This paper was presented as "The Dual Solution System—A Different Approach to Some of the Problems of Air Conditioning" at the ASHRAE 67th Annual Meeting, Vancouver, B. C., June 13-15, 1960.

solution when cooling) is now recirculated directly between the oil fired first evaporator and the solution cooler. Humidification is accomplished with same pumps and sprays used with the primary solution when cooling.

The heating and humidifying cycle is shown in Fig. 2.

Efficiency and Performance – The primary justifications for considering the dual solution system are its low operating cost and independent control of humidity and temperature within limits.

To establish the potentialities with regard to operating cost, a representative set of conditions has been established and the predicted temperatures and quantities throughout the system indicated in Fig. 3. The following assumptions were made.

60,000 Btu/hr removed from air by reducing it from 80 db 68.8 F wb (50% rh 60 F dp) to 65 db 60.7 F wb (78% rh 58 F dp).

Heat removed from secondary solution by cooling it from 95 to 83 F using outdoor air at 95 db 75 F wb water saturated.

Oil or gas fired first effect evaporator – 75% furnace off.

These will be recognized as moderate design specifications for a home air conditioner. They will be used to indicate normal maximum fuel and power requirements which can be compared with conventional equipment of similar capacity. The possibilities of the dual solution system in meeting more unusual conditions will then be explored.

Quantities and temperatures to be expected are indicated in Fig. 3. Demonstration of the entire heat balance is complex and believed to be unnecessary for the present purposes, but some calculations are pertinent to justifying the quantities of Fig. 3 as follows:

1—Water absorbed by secondary solution.

a) Equiv. of 60,000 Btu/hr $= \frac{60{,}000}{1060} = 56.5$ lb/hr

b) Loss due to air recirculation in vapor exchanger. Approx. 3300 cfm at 2.5 in. Abs cooled from 85 to 65 F = 5800 Btu/hr equiv to 5.5 additional lb/hr water

c) Total absorbed by secondary = 56.5 + 5.5 = 62.0 lb/hr

2—Quantities fed to concentration system

Secondary solution strength 36 %
Returning concentrated solution 41 % (limited to avoid solidification during shutdown)
X = lb/hr LiCl in feed
Y = lb/hr H_2O in feed

$$\frac{X}{X+Y} = 0.36$$

$$\frac{X}{X+(Y-62.0)} = 0.41$$

X = 186 lb/hr
Y = 330 lb/hr
Feed = 516 lb/hr

3—Evaporation in each effect

a) Latent heats of evaporation

	First effect	Second effect
Vacuum	10 in.	27.5 in.
Solution temperature	260 F	170 F
Heats Btu/hr/lb		
Evaporation—water from water	983	1033
Heat of solution approx.	80	80
Heat to superheat vapor—approx.	33	35
Total latent heat evap. fr. solution	1096	1148
b) Latent heats of condensation	1016	1068

c) Each pound of vapor evaporated in the first effect will (neglecting losses) condense and thereby evaporate $\frac{1016}{1148} =$ 0.89 lb in the second effect. Neglect of losses in this case is justified by the experimental observation that feed to the second effect is likely to be superheated and hence flashes off some vapor.

4—Required heat input to first effect

a) Heat to feed in first effect. In the experimental unit feed was found to be approximately 20 F below evaporating temperature. This approach could be considerably improved by insulation etc. but is assumed for these calculations

Specific heat of strong LiCl solution is approximately 0.6
First effect heat to feed = 516 × 20 × 0.6 = 6,200 Btu/hr

b) Required evaporation in first effect $= \frac{1.0}{1.89} \times 62.0 = 32.8$ lb/hr

c) Total net input to first evaporator = 32.8 × 1096 + 6,200 = 42,100 Btu/hr

5—Heat input to burner $= \frac{42{,}100}{0.75} =$ 56,200 Btu/hr

6—Heat rejected to outdoor air = 60,000 + 42,100 = 102,100 Btu/hr

7—Auxiliary power

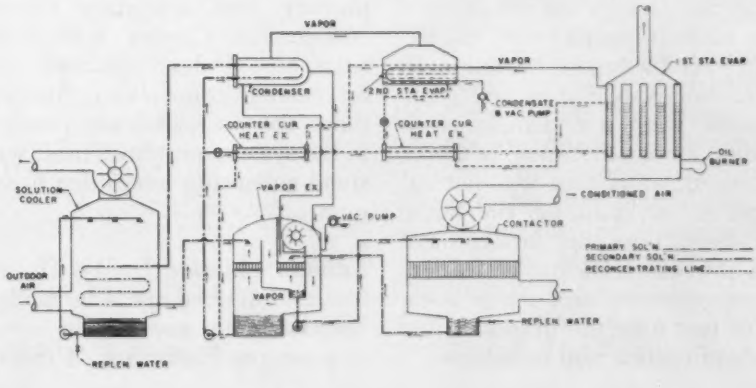
Small pumps operating continuously require an aggregate of about 0.4 hp. Vacuum pump drive ¼ hp intermittent. Vapor exchanger fan less than 0.07/hp continuous.

The foregoing has been assembled for a 5 ton (60,000 Btu/hr) system as this was the design capacity of the experimental unit, and in general, the experimental results support this analysis although no attempt has yet been made at an accurate heat balance or other performance test. The above figures can be summarized as follows:

Fuel input to burner Btu/hr per ton refrig.	11,200
Heat to outdoor air Btu/hr per ton refrig.	20,400
Auxiliary power hp per ton refrig.	0.1

If these performance figures are translated into operating costs using normal fuel and electrical energy costs and then compared with any of the conventional systems, it will be found that a great advantage appears to be available.

Fig. 1 Arrangement of the double effect evaporator system



The second outstanding asset of the dual solution is its ability independently to control moisture content and temperature of conditioned air. This control is achieved through the primary solution. As has been shown, the water evaporated from the primary solution in the vapor exchanger accounts for the entire refrigerating load—both sensible and latent – plus some losses. Consequently, if no make-up is provided the primary solution will tend to become more and more concentrated.

As the limit on evaporation from the primary solution is a function of the vapor pressure maintained by the secondary solution in the vapor exchanger, the effect of concentrating the primary solution will also be to allow its temperature to rise. The end result would then be that the primary solution would greatly increase its capacity for absorbing moisture at the expense of its ability to remove sensible heat.

To control the proportions of sensible and latent heat removed from the air, the primary solution is rediluted continuously by controlled make-up water. It is this arrangement that makes it possible and practical to adjust to suit requirements. Table I shows the approximate desired strength and temperature of the primary solution for several assumed load conditions.

The values in Table I are illustrative but would actually depend upon the performance of the contactor and quantities of air and solution being circulated. The table is intended to indicate that the system has an inherent capacity to adjust itself to a wide range of requirements.

TABLE I

General Condition	Design Maximum Day	Light Load	High Moisture Load
Return Air			
Temp db	80	75	80
Rel. Hum.	50%	50%	50%
Supply Air			
Temp db	65	70	70
Dew Point	58	55.5	55
Prim. Soln.			
Temp.	58	65	65
Strength	13%	21%	23%
Equiv. Dew Point	53	52	50

The Experimental Unit – Figs. 1 and 2 show the experimental unit as set up in the author's home. The supply and return air connections were made in parallel with the existing forced warm air heating unit. As the house duct sizes and grille locations were not planned with summer air conditioning in mind, the experimental unit was not expected to air condition the house effectively but the arrangement does provide a means of loading the experimental unit and it is expected that some useful cooling and dehumidification will be secured.

The experimental unit was designed for five ton capacity when maintaining 80 F, 50% rh with outdoor conditions of 95 db and 75 F wb. It now appears that its capacity will be limited to somewhat less by one or two component parts, unless they are redesigned and replaced.

The vapor exchanger is fabricated from steel and fitted with two sets of sprays, contact material, and a motor driven fan. Epoxy and synthetic rubber coatings have been successful deterrents of corrosion.

The first effect evaporator uses a low pressure type oil burner. All parts in contact with solution are silicon bronze with welded or silver soldered joints.

The condenser and heat exchangers are copper throughout with brazed or silver soldered joints. The means of treating the air to be conditioned (the "contactor") is a substantially standard capillary air washer with sprays and glass fibre contact cells. The solution cooler is a standard evaporative condenser. The main solution pumps are centrifugal. Small positive displacement pumps control flow in the concentration circuit. The vacuum pump is of the sliding vane type with its individual motor but the other pumps are driven by a single motor.

Hose was used for the main primary and secondary solution connections. Copper tubing and brass pipe and fittings were used for smaller connections. Although there are no significant pressures in the system most lines must withstand collapsing forces due to vacuum.

Lithium Chloride Used – The system as described could be made to function with any one of several hygroscopic solutions. However, lithium chloride was chosen for several reasons for both primary and secondary solutions. Lithium chloride has been widely used for conditioning air for many years. It has been applied for human comfort and for industrial purposes. It has had no deleterious effect on either humans or products and in fact is said to have beneficial germicidal properties.

Among the hygroscopic liquids it is especially attractive because it retains reasonably good flow characteristics even at high concentrations and because, although an electrolyte, its corrosive tendencies have been found to be controllable. Lithium bromide, a similar material, has been favored for sealed absorption systems presumably because of a better corrosion behavior.

Installation of the equipment was completed in 1954. Trials, tests and development have continued since then. From the outset it was recognized that a primary objective was to achieve substantially automatic functioning of the entire system including the double effect evaporator. Furthermore, there must be reasonable assurance of freedom from mechanical or corrosive failure. For these reasons, rather costly materials and fabrication methods were used, with the hope and expectation that substitutions could be made when the basic system had been proven. The satisfactory use of lithium chloride in such a system could not be taken for granted.

CONTROLS

Automatic control of the unit was provided even where it resulted in complexity incommensurate with the capacity of the unit. It was felt to be important to first determine whether the relatively complex system was amenable to automatic operation and to thereafter seek ways to reduce cost and complexity.

The first stage evaporator temperature is maintained by aquastat for either cooling or heating conditions. For the cooling cycle, vacuum is normally maintained in the vapor exchanger continuously. Upon demand for cooling the vapor exchanger fan and pumps are started, setting the cycle in motion. The conditioned air fan may be

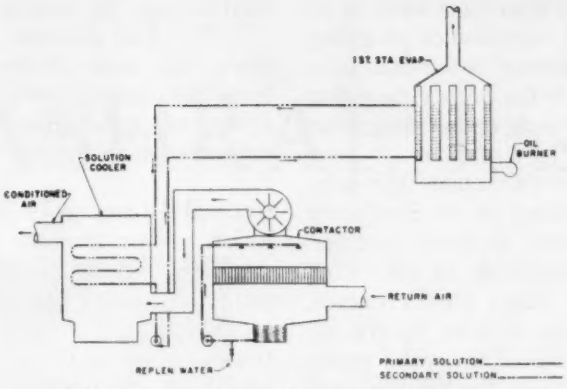


Fig. 2 Heating and humidifying cycle

run continuously or not as desired. Careful attention to gravity flow and air entrapment, where necessary, have resulted in quite satisfactory automatic operation of the evaporator circuit.

Although no such controls were installed, it is apparent that by regulating water return to the primary solution, (for which water loss by evaporating corresponds to the total refrigerating load) its strength and resultant dehumidifying capacity can be controlled over fairly wide limits. Its temperature can be controlled by conventional bypassing.

Safety controls consist of the usual ones on the first evaporator plus a low vacuum cutout of pumps, responsive to vapor exchanger vacuum.

The experimental unit has not been equipped to convert quickly from cooling to heating or vice versa although it has functioned successfully in both capacities. Provisions for rapid and easy conversion of this unit would entail rather elaborate motorized valve and damper linkages. Much could undoubtedly be accomplished with further testing and design studies but it may well prove to be a rather costly provision in a small system.

Test Results and Limitations – As stated, the primary objective of these first tests was to establish feasibility. No attempt was made to achieve optimum conditioned air temperature or humidity. To do so, it was soon apparent that a major rearrangement of the internal structure of the vapor exchanger would be required. However, enough experience was gained to indicate what was possible and practical.

The dual solution system, since it interposes another solution and an inert vacuum air stream between the strong solution and the air being conditioned, limits the obtainable moisture reduction for a given strong solution strength and temperature. Furthermore, it imposes practical limits on the strength of the secondary solution, since it must be restorable to required strength by extracting and further concentrating a small portion, without that portion being concentrated enough to solidify and plug the system during shutdown.

There are no precise limits to obtainable temperatures and humidities as they are influenced by design and acceptable losses in efficiency. The tests indicated that the dual solution system is quite able to satisfy usual requirements for human comfort air conditioning when the secondary solution is cooled to temperatures obtainable from evaporative condenser-like equipment supplied with saturated air at 80 F wet bulb or less.

It is significant that because the dual solution system controls temperature and humidity independently and without "bypassing" of air it should achieve a better control of humidity within its limits than do other systems. This is especially true in comparison with direct expansion fluorocarbon or sealed lithium bromide types.

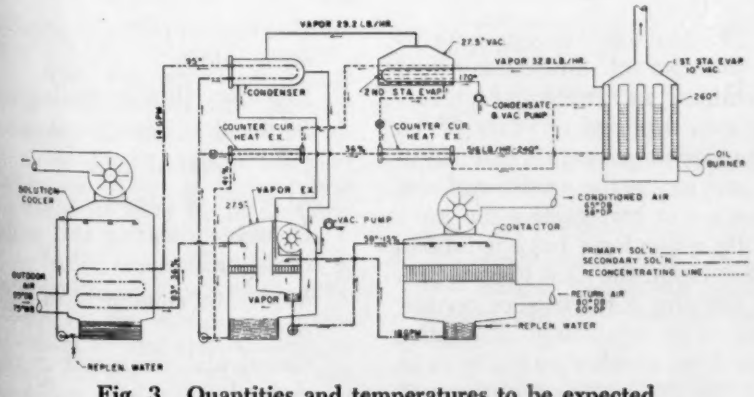


Fig. 3 Quantities and temperatures to be expected

Disadvantages – The outstanding disadvantage of the dual system should be recognized at the outset; probably inherent, it would be a vital factor in establishing applicability. Many parts are required; some of which are bulky. This implies added first cost and added space requirements.

As compared to compression refrigeration, the heat rejection is considerably higher. For this reason straight water cooling is not proposed for normal applications of the dual solution system. It might be attractive where there is ample water, such as sea water, available. The system as herein described requires only a little more make-up water than a compression unit served by a cooling tower but it must have that much water. The arrangement shown cannot operate on dry air cooling under normal conditions.

There are many applications of air conditioning, especially in vehicles, where pumps and sprays are not acceptable. For such applications, direct expansion compression refrigeration is about the only answer.

As compared with presently used open absorption systems it lacks the ability to achieve extremely low humidities sometimes required in process industries. These rather low humidities are only obtainable with any system at relatively high first cost and operating cost.

Advantages—The disadvantages are

serious and assuredly would prevent any considerable application of the dual solution system were there not compensating advantages. It does have many potential advantages but probably no one of them is outstanding,

First, consider comparison with compression refrigeration, specifically of the direct expansion type including the heat pump. The dual solution system provides independent control within its capacity. It is able to take care of that difficult condition of high latent and low sensible heat loads on the "muggy" days. It avoids the difficulties of wet cooling coils after compressor cut off on intermittent operation.

Relative operating costs are dependent on fuel and electric power costs, but it appears that the dual solution system will have a large advantage under usual conditions. Gas utilities and the fuel oil industry are known to be seeking more summer load and the dual solution system offers one approach. An important asset of the dual solution system is that it does not require a large increase in electrical service or consumption. This can be quite important where either wiring or generating capacity is limited. It might make quite a difference in some of the undeveloped areas near the oil fields.

In the combined heating and cooling configuration the dual solution system, as does the heat pump, utilizes the same equipment for heating and for cooling. An advantage over the heat pump, however, is that there need be no limitation on heating capacity with reference to cooling capacity. This means that the system can be applied economically in localities or under conditions where the heating load may be expected to be large compared to the cooling load.

Consider the dual solution system in comparison with other open absorption systems. An outstanding asset is that it avoids heating and evaporation from an aerated and contaminated solution. Experience with the experimental unit indicates this to be quite significant. The strong secondary solution recirculates within a clean system and is deaerated continuously in the vapor exchanger. The weak primary solution is never heated. After several years of intermittent experimental operation there is no indication of corrosion or plugging.

The efficiency of the dual solution system is far higher than that of conventional open absorption systems. It is customary in such systems to reconcentrate the solution by causing it to flow over heated contact material countercurrent to a stream of air. This, aside from other inefficiencies, causes a loss of heat to the air stream carrying off the water vapor.

Furthermore, the solution used for air treatment cannot, without the use of refrigeration, be colder than the available cooling water. Under normal conditions this results in heating the air while drying it. The open absorption system as just described does not take advantage of multiple effect evaporation and cannot readily do so.

The dual solution system permits treating the air with a primary solution of different composition than that of the high strength secondary solution. Thus, the solution used to treat the air may have special germicidal or other properties but need not be highly hygroscopic nor need it be exposed to heat. This indicates possibilities in some special applications such as in hospitals. In practice the two solutions must be reasonably compatible as there is likely to be some carry-over in the vapor exchanger.

Finally, consider the advantages of the dual solution system compared with sealed heat powered systems such as those currently in use employing lithium bromide. The latter systems avoid moving parts but are thereby denied the advantages of multiple effect evaporation and forced flow through heat exchangers. The consequence is a large difference in efficiency in favor of the dual solution system.

The dual solution system avoids the problem of establishing and maintaining an extremely high vacuum over a period of years. Even minor internal corrosion can add to this problem as the products of corrosion are in part gaseous.

By achieving a large reduction in heat input, the dual solution system requires a far smaller cooling tower or its equivalent. If cooling water from another source is to be used, the dual solution system will then require far less of the water.

The dual solution system has about the same advantages as to humidity control over the sealed absorption systems as it does over compression refrigeration.

General Advantages – The dual solution system imposes no unusual standards of workmanship or material. The experimental unit was assembled with no unusual precautions as to tightness. Standard plumbing and tubing fittings were used in making connections. It has been taken apart and reassembled many times without difficulty.

In general comparison with other systems including heating systems it is significant that the dual solution system employs no pressures at any time and that there is no hazard from rupture. The liquid and vacuum pumps operate at low heads and moderate vacuums.

As to freedom from noise and vibration, the dual solution system is almost, but not quite, as favorably endowed as the sealed absorption systems. It is at a definite advantage when compared with the other systems.

CONCLUSIONS

The dual solution system cannot be expected to find a place among room units or small simple systems where size and cost are paramount considerations. There are many transportation and similar applications for which it is unsuited. It lacks the ability to produce extremely low humidities needed for some processes. The most attractive possibilities are probably for those somewhat unusual human comfort applications where, for one or more reasons, we can accept a degree of complexity in return for:

- Unusual control of conditioned air humidity
- Efficiency (low operating cost)
- Reduced vibration and noise
- Low electrical load
- Absence of pressure
- Combined heating and cooling if required
- Moderate standards of tightness
- Germicidal treatment of air if desired

ASHRAE has problems

let's face them squarely

WALTER A. GRANT
President
ASHRAE

Eighteen months have passed since the merger of ASHAE and ASRE. It is time for us take stock of how we have done, how we're doing today, and what lies along the road ahead.

Our Society has come a long way in eighteen months. Once the merger vote was taken, the membership at both chapter and national levels closed ranks and resolved to make it work.

Nevertheless, we are confronted with problems, a few of which are serious, and many of which are difficult and complicated. All of the important ones had their roots in the predecessor organizations long before the combination.

In the months to come, your officer group proposes to discuss these problems frankly with you, rather than yield to the temptation to look the other way, and hope that the problems will solve themselves and go away.

We propose to face them squarely, and with your help determine the best possible solutions.

The basic, overall question is simply stated: Is your management—technical and non-technical, at all levels, nationally and locally—advancing the wishes of, and the best long-range interests of, the membership?

While experienced leaders in any organization necessarily will be buffeted by the eddies and cross-currents of events and opinions, in the long run they are obliged to determine a wise and consistent course in tune with the membership in order to make constructive progress.

From this basic question we come to an examination of each of our problem areas.

We have a financial problem which cannot be swept under the rug. I might add that so has every other technical society today—we're not unique and we're in good company. During the coming fiscal year we will give every member $60 value in services for his $25 annual dues. This is much better value than most organizations give. But to sustain our source of income supplementary to dues requires a tremendous effort on the part of many devoted members. What services in the way of publications, long-range research, meetings, and expositions do our members really want to continue and support during the years to come?

The Society By-Laws have built-in safeguards to assure adequate representation of and service to the three major areas of membership interest, namely (1) heating, (2) refrigeration, and (3) air conditioning and ventilation. A review of meeting programs, publications and committee memberships will show that each of these broad areas is, in actual fact, very equitably represented and actively served. But have we set up adequate machinery and put the wheels in motion to satisfy the needs of particular segments of these broad areas? What about cryogenics? Air pollution? Food processing and storage? Drying of materials? What about programming at the chapter level to satisfy specialized groups and simultaneously maintain the interest of the average member? While the problems of special-interest groups existed long before the merger, it remains our responsibility to find satisfactory solutions.

In the area of publications, we have several problems. Precisely what policy goals should we establish for the JOURNAL in order to make it one of the outstanding engineering publications in the country? Do our members want to continue the Society Transactions? How well are the Guide and Data Book and the various miscellaneous publications serving you?

In our Research Laboratory at Cleveland, we possess an organization which is unique among engineering societies, and which has made notable contributions in the past. In what directions should it continue? Should it be expanded or contracted? These are questions not to be answered lightly on the basis of personal opinion, but only after the most searching examination of our long-range policy and objectives.

Finally, is our organization structure suited to the needs of a fast-moving engineering society? Are we not, perhaps, "committee happy"? Shouldn't we take a look in the mirror at our administrative and committee organizations to be sure they are serving us as effectively as possible?

In the months to come, your officers, Board of Directors, committees at interest, and Headquarters Staff will be devoting much time and effort to bring about wise solutions to these problems. These people are determined to respond to the membership, and at the same time maintain the high standards expected of an engineering society. We need your active interest and your constructive suggestions to help us formulate the long-range goals and programs needed for our Society's future growth.

. . . $60 value in services for $25 annual dues . . .

. . . what services do members really want . . .

. . . how to make the JOURNAL one of the outstanding engineering publications . . .

. . . should our Research Laboratory, unique among engineering societies, be expanded or contracted . . .

. . . are we "committee happy"

A simple solution to completely automatic change-overs in air conditioning is the Three-pipe System

A. I. McFARLAN
Member ASHRAE

There have been a number of conflicting and sometimes confusing statements regarding the application of three-pipe systems to air conditioned buildings. Some of these statements are correct for one application, and entirely incorrect for another. The purpose of the data submitted below is intended to be more or less basic, which when analyzed, will be helpful in choosing between the customary two-pipe system and a three-pipe system. If the three-pipe system is applicable and economically feasible, then there are many factors which, if properly considered, can be the means of a three-pipe system working most economically. Equipment selection, particularly the air distribution, water quantities circulated, design of coil surface, number of stages used in the refrigeration system, as well as the choice of prime movers all have an important bearing on the first cost and most economic application of three-pipe systems.

Fig. 1 shows a hypothetical cycle. The chilled water from the water cooler and the warm condensing water are mixed in controlled quantities to produce a controlled temperature T-4, which is the result of the mixture. Water at temperature T-4 returns to both the water cooler and the condenser. If such a cycle were run over a period of time, the condensing water temperature would build up due to the addition of the compressor heat.

By installing a second heat exchanger after the condenser, and keeping T-3 controlled at a set point, then the heat removed from the second heat exchanger is the quantity of heat radiated from the chilled water cycle plus the heat added by the compressor, plus the heat added from the circulating pump.

To illustrate the point, neglecting radiation, if the temperature leaving the water cooler (T-1) were 45 F, the temperature leaving the condenser (T-2) were 95 F, and the heat removed by the water cooler were exactly 100 ton, then the compressor hp would be approximately 100 hp. If the circulating pump required 5 hp, the heat that would have to be removed by the second heat exchanger in order to keep the circuit in balance would have to be

$$\frac{(100 \text{ hp} + 5 \text{ hp}) \times 2545 \text{ Btu/hp}}{12{,}000 \text{ Btu/ton}} = 22.3 \text{ ton}$$

Forgetting for a moment the pump hp which must be used in any similar cycle, the ratio of the excess heat removed from the cycle is approximately in the ratio of 1 to 5, compared with the heat removed by the water cooler. This is an academic analysis of the customary peak load performance for an air conditioning system using single stage compression in the range of temperatures selected.

Fig. 2 is an oversimplified curve which will more or less apply to different types of refrigeration systems. This curve is offered only for point of illustration in order to generalize. Actually, for the correct analysis, the curve for the particular type of equipment under consideration would have to be analyzed.

If the refrigerating equipment were operated at no load, an amount of power would be consumed depending upon the type of equipment, but shown here as 20% of full load. In actual performance, many variable factors would alter this figure, such as quantity of surface used, whether absorption, steam jet, electric driven compression, or steam driven compression were selected. This also assumes peak load conditions approximately as outlined above.

"In the three-pipe system we have a coil which can add heat to the cold line and a second coil which can remove heat from the hot line. . . . The closer the temperatures can be maintained between the hot and cold line, the less this heat and cooling will cost. . . . Each coil is provided with a three-way valve to mix two water temperatures to provide the exact temperatures of hot air and cold air required at any time. . . . This cycle has been in operation in a number of projects for over five years and is a simple solution to avoid the inconvenience of change-overs, and at the same time transfer heat from one part of the building to another, completely automatically."

A. I. McFarlan is President, A. I. McFarlan Company, Inc. This paper was presented as "Basic Data for Analysis in Application of Three-Pipe Systems", at ASHRAE Chapter Meeting in Toronto, Ontario, April 4, 1960.

Re-examining Fig. 1, if instead of 45 F leaving the water cooler the thermostat is reset at T-1 for a temperature only a few degrees lower than T-3, then again the compressor would unload and the heat removed by the second heat exchanger would be much less than the case cited above, where T-1 was 45 and T-2 was 95 F. The system can be unloaded if the chilled water and condensing water temperatures are brought closer together.

Since less heat is removed from the second heat exchanger, the power falls off approaching the low side of the curve in Fig. 2.

If this cycle were flexible, the condenser and water cooler surface could be expanded, simultaneously increasing the amount of water circulated. If these three variables were to be increased with T-1 and T-4 controlled but only a few degrees apart until the compressor was fully loaded, then the capacity of the system would greatly increase above 100 ton and the former 1-5 ratio would also increase substantially.

Taking another condition of 55 F chilled water (T-1) and the mixed water (T-4) equal to 65 F. If the heat transfer surface is worked out to load the compressor and enough water is circulated for a range of about one-third of the full load range, then the performance of a 3-stage water cooling cycle operating on one of three equal stages would be duplicated with surface to dissipate the heat under operating conditions between seasons with wide range coils. The application of this cycle is discussed further.

As the spread between the suction temperature and condensing temperature of any refrigerating cycle is reduced, regardless of equipment, the capacity of the cycle is increased and the power required to operate the cycle is decreased.

This assumes that the additional surface is put into the water cooler and condenser to take advantage of the additional capacity.

A basic conclusion drawn from the analysis thus far: When increasing the temperature of water leaving the water cooler, and simultaneously dropping the condensing temperature, the power will vary somewhere between 20% and 100% of the full load power, depending upon the temperatures operated. To illustrate this: If the refrigerating machine is kept running but set T-1 equal to T-3, then approximately 20% of full load power would be used. As the differences are spread between T-1 and T-3, up to 100% power for the peak design conditions already mentioned are used.

THREE-PIPE SYSTEM

Fig. 3 shows a three-pipe system which is substantially the same as the cycle shown in Fig. 1, except there is a coil which can add heat to the cold line and a second coil which can remove heat from the hot line (derived from the condenser).

If the air quantities across both coils are controlled, the system can be kept in balance exactly as was the case in Fig. 1.

The same conclusion can be reached. If the temperature of the air is the same entering both coils, then the closer the temperatures are maintained in the hot line and cold line, the less power will be required.

Other conclusions are as follows:

1. Any time that air can be cooled during the heating season, the amount of heat in the form of hot air equal to the heat removed from the cooled air is returned, plus an amount which would have been removed by the second heat exchanger in Fig. 1.

2. The closer the temperatures can be maintained between the hot line and the cold line, the less this heat and cooling will cost.

3. Since all of the heat removed from the water cooler is returned (five parts to the addition of the compressor heat in the relation of one part), roughly six parts of hot air is secured to the one part which represents the compressor heat for operating conditions equivalent to the full load summer conditions. As the hot and cold water temperatures more closely approach each other, the 6 to 1 ratio increases substantially.

4. Typical entering air temperatures for peak summer conditions should be somewhere between 84 and 95 F depending upon the percentage of outside air. During the heating season there would be temperatures entering the coils below the room temperature when mixed with outside air. Obviously, the approach temperatures of the hot line and cold line will be much closer for between season conditions.

5. If a ratio of 1.2 of heated air load to cold air load can always be obtained, then the cost of our heat transferred within the building will be not more than 1/6 of the cost of direct electric heat. Here again, the cost of the power for the circulating pumps is eliminated, since most large air conditioning systems utilize pumps for hot water in the win-

Fig. 1 Basic cycle for analysis only

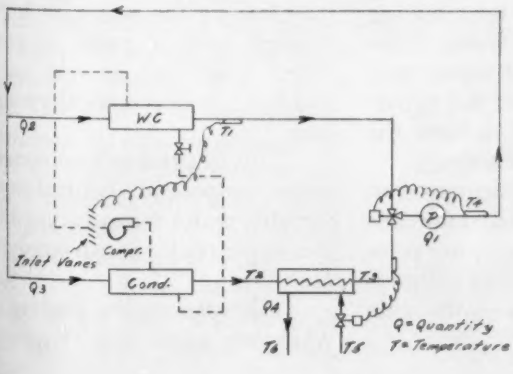
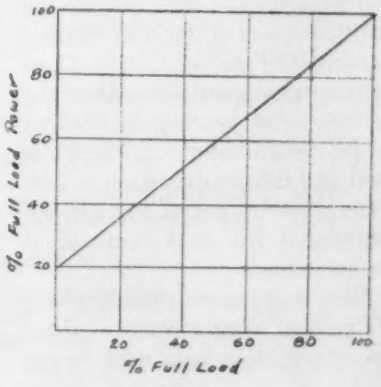


Fig. 2 Typical curve of refrigerating equipment used for air conditioning



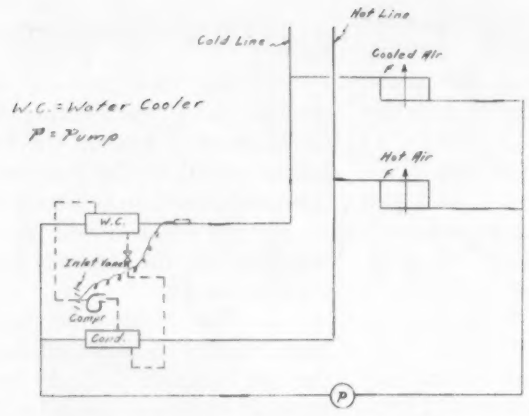


Fig. 3 Three-pipe system with coil that can add heat to the cold line and a coil which can remove heat from the hot line; the heat coming from the condenser

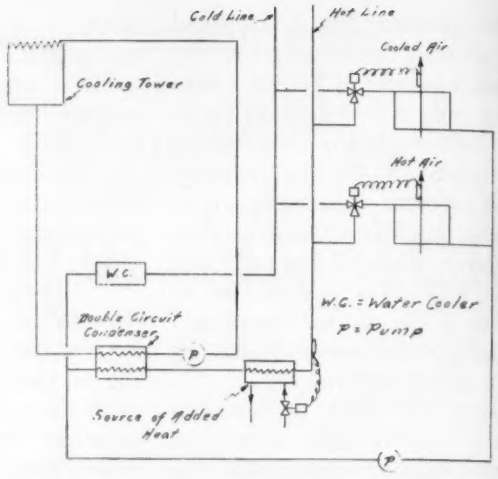


Fig. 4 Typical three-pipe closed cycle system

ter, and this is a relatively small item in the total overall comparison. This heat is returned in the form of useful heat, but at a cost more nearly approaching the cost of direct electric heat.

CLOSED CYCLE

Fig. 4 shows a typical three-pipe Closed Cycle system. There is added to Fig. 3, a double condenser and cooling tower to remove excess heat, and a converter to make up a deficiency of heat. Each coil is provided with a three-way valve to mix two water temperatures to provide the exact temperatures of hot air and cold air required at any time.

This cycle has been in operation in a number of projects for over five years, and is a simple solution to avoid the inconvenience of change-overs, and at the same time transfer heat from one part of the building to another, completely automatically.

For simplification, there is shown only one stage which is the stage equipped with the double condenser. Generally one of three equal stages will carry the in-between season load with outside air quantities selected for the normal peak summer load ventilation. Contrary to common opinion, where the 6 to 5 ratio of heating to cooling can be maintained, nothing is gained and frequently much is lost, if more than required outside air is circulated for heat removal in between seasons.

Note that the circulating pump now passes approximately three times the gallons per minute per ton (assuming two stages inoperative). Yet, if the coils are designed for 24 F rise at full load, rather than the usual 5 to 12 F rise, then the mains can be installed ¼ to ½ the size of conventional methods. Since these mains are insulated, this results in a large first cost savings.

The same coils can now be used for heating and cooling. Likewise, the same but much smaller pumps and controls (due to handling less water at full load) all contribute substantial reductions to first cost.

If this system could be designed exactly for full load, the coils all would be passing 100% cold water. If, however, there is a safety factor in some coils, or with zoning due to sun effect on one side of the building and not on the other, the control valve for the oversized coil will cause some reduction of cold water, and will pass some water from the so-called hot line. During peak summer loads there is no heat added from the converter or second exchanger, so that any heat added can come only from the double condenser circuit. Any such heat removed from the condenser has already reduced the condensing temperature of the refrigerating machine if the return water is below the cooling tower water temperature. This increases the capacity of the system and reduces the hp, as indicated previously.

On a centrifugal compressor, it does not make much difference if the condensing temperature is reduced or the suction temperature is increased. In either case, the gain is almost the same.

On reciprocating compressors there is much more gain in capacity by increasing suction temperature. There is a gain, however, in hp by either increasing the suction temperature or decreasing the condensing temperature.

In other forms of refrigerating equipment, the general principle of reducing condensing temperature or increasing suction temperature, will in varying amounts increase capacity and decrease the power necessary to operate the specific system.

When heating and cooling are required simultaneously, it is particularly desirable in a three-pipe system to automatically raise the temperature of the water in the cold line to a maximum and decrease the temperature of the water in the hot line to a minimum. This is easily accomplished by extending the operation of the control valve on each coil. When wide open on the cooling side and still not satisfying the temperature of the thermostat in the leaving air stream, then the temperature of the water leaving the water cooler is reduced. As long as all thermostats in the air stream are satisfied, then the water temperature leaving the water cooler is increased until a point is reached where one coil is not satisfied, which again decreases the temperature.

This is a rather new concept in water temperature control in that a variable water temperature leaving the water cooler is extremely desirable.

Likewise, on the heating cycle, whenever a coil requiring heat has

opened the control valve wide on the heating side, and still is not satisfying the thermostat, the valve continues to turn, which opens the steam valve on the converter. When the water temperature has been boosted too much, then the three-way valves will start to close, which in turn, drops the hot line temperature by throttling the quantity of steam to the booster converter.

This is the solution to the requirements developed previously, in maintaining automatically the cold line temperature to a maximum, and the hot line temperature to a minimum.

At the same time, this provides a means of automatically removing the excess heat through the cooling tower, and utilizing the heat from the condenser for temperature control, or as a heat source any time of the year.

Whenever heat is utilized during the cooling season, the condensing temperature is dropped below what it would be with the cooling tower alone, so that the utilization of such heat is actually a gain, even during the cooling season.

The closer the approach of the entering water to the leaving air, and the closer the approach of the leaving water to the entering air, the closer will be the temperatures in the cold line and the hot line, which is important in a three-pipe system.

COIL DESIGN

Under present design standards, it is not uncommon to cool all outside air from 95 F with a water rise across the coil in the magnitude of 10 F, for instance, 42 to 52 F. It is possible for a relatively low increase in coil cost to take the water temperature much higher for design conditions, and this additional surface pays off to a marked extent in a three-pipe Closed Cycle.

To some extent, the wide range coil is desirable on any single stage refrigeration system, but on a Staged Cycle, it is a prerequisite. Compressor hp can be reduced 25 to 40 % by staging compressors and properly selecting wide water range coils. This results in a year round gain on a three-pipe Staged & Closed Cycle system, at the same time reducing the first cost of mains, insulation, pumps, motors, control valves and coil connections, as well.

Analysis so far has applied only to the Closed Cycle, i.e., where the condenser heat is utilized as a source of heat in the air conditioning system. In the Closed Cycle, the mixing of two temperatures of water serves many purposes and produces economic gains.

Conversely, however, where the heat source is not from the cycle itself, but, for instance, entirely from a boiler or converter, and the cooling source is from a water cooler, then mixing of hot water and cold water results in one nullifying the other.

The general statement that mixing is either feasible or unfeasible, wasteful or economical, is meaningless unless the type of system is defined. What is wasteful for one type of system is an economic gain in the other, in addition to being a solution to temperature and humidity control, with one point of control for heating and cooling, and one coil used for both heating and cooling.

SPACE SAVING

In choosing a three-pipe or two-pipe system, one of the major factors in addition to economy, is the space savings made possible by the elimination of double fans. This in turn results from a system which uses a constant quantity of ventilating air the year around, providing heating or cooling without changing the outside air quantity. There is a common misconception that outside air is the cheapest way to cool the interior of a building during the heating season. This frequently is not the case. It is generally much cheaper to transfer heat than to generate it. In a building with a large interior load, the economics will usually justify the three-pipe Closed Cycle. Also, space saving alone may justify the three-pipe Closed Cycle. So the gain is doubled.

To illustrate this point, installed and analyzed were several other multi-story office buildings where a low pressure single fan system can be installed for the core area in less space than is required for the air duct shafts alone. This may require combining several floors on one system, but this is no different than the commonly used high velocity systems where ten to twenty floors are combined on one core system. A perceptible gain in

TABLE I

TYPICAL BOILER HEAT COSTS

Basis: Oil, electricity, labor		$.10/gal.
Boiler efficiency		70%
Standby & pipe losses between seasons		20%
140,000 Btu/gal. oil		
140,000 × .70 × .80 = 78,400 Btu	costs	$.10
100,000 Btu's	cost	.128
Direct electric @ $.01 per kw-hr		
Assumed energy charge		$.007
Assumed demand charge		.003
Including demand	Energy charge alone	
3,450 Btu cost $.01		= .007
100,000 Btu cost .29		= .202

TYPICAL CLOSED CYCLE HEAT COST

(Transfer of heat requires maximum of 60 hp per 100 ton)

100 ton	=	1,200,000 Btu
60 hp × 2545	=	152,000
Recovered heat		1,352,000

At $.01 per kw-hr		At $.007 energy charge	
$60 \text{ hp} \times \frac{.746}{85\% \text{ eff.}} \times .01 = \$.525$		$60 \times \frac{.746}{.85} \times .007 = \$.369$	
1,352,000 Btu cost	.525	1,352,000 Btu cost	.369
100,000 Btu cost	.039	100,000 Btu cost	.027

SUMMARY

Boiler heat, cost per 100,000 Btu (between seasons)	=	$.128
Direct electric heat, including demand	=	.29
Direct electric heat energy charge only	=	.202
Closed cycle, including demand	=	.039
Closed cycle energy charge only	=	.027

space is made possible by the elimination of double fans, multiple dampers, and by the use of a constant quantity of ventilating air the year around.

In many cases it is possible to use two air streams, one for the interior and one for the exterior, and eliminate expensive exterior high pressure risers, induction units and unit enclosures. Consideration of this type system is particularly recommended for the cantilever construction office building where deep beams between floors leave ample space for running ducts.

Double duct systems present all kinds of possibilities in using all coils for both cooling and heating. Again, the possible transfer of heat from interior to exterior makes double fans a liability.

TYPICAL FUEL COST ANALYSIS

Here is a cost analysis based upon transferring heat with electricity as a prime mover on the refrigeration system compared with using oil as a fuel source.

In Toronto electricity in large quantities costs about one cent a kilowatt and fuel cost is about ten cents per gallon for the equivalent of a United States gallon of No. 6 (Bunker C) oil. By using these units, a correction is readily possible, viz. for electricity at 1.2 c per kw, multiply the costs by 1.2

On this basis, the relative costs are those as shown in Table I.

It is obvious that with a reasonably favorable electric rate, it is cheaper to transfer heat using 60 hp per 100 ton for this transfer, which figure has been well established over five years' operation. While this transfer would occur mainly during the period when lights were on, and would have to be computed at the electric cost including demand charge, there are many times when the transfer can be accomplished at less than peak demand. Again, this requires an accurate survey of each possibility.

ANALYSIS CONSIDERATIONS

One of the first factors to be considered in a practical analysis of any large project is the heat that must be removed from the core area during the heating season. In general, most buildings have a considerably greater cooling load for the periphery than for the core. During the heating season, it will be the core area which will require the cooling, and the periphery which will require heating. When all or part of the periphery is taken off, the cooling load required for the winter season is substantially reduced. In addition, all of the outside air load will have been removed, since the outside temperature is below the inside temperature, and always has some cooling available from the ventilation air.

There will be a break even point in all buildings when the outside temperature drops sufficiently low that the ventilating air can carry the cooling requirements. This will vary with each building, but it is reasonable to say that whenever heating and cooling are required simultaneously chilled water temperatures well above the summer requirements are possible. Likewise, hot water temperatures well below the peak winter requirements are also possible. Whether hot water has entered a wide range coil with substantially reduced load for air heating, or if cold water has entered another coil for air cooling, the outlet temperature will be nearly the same. All of these natural results follow the basic requirement for economic operation of a three-pipe Closed Cycle system.

If wide range coils are used and a Staged Cycle which necessarily requires the full tonnage condenser and water cooler surface, but smaller compressors and motors, almost every factor conducive to producing a minimum temperature difference between hot water and cold water will be obtained, at the same time using not more than one-third of the refrigerating plant power for winter operation, assuming winter operation with one stage of a three stage system.

Summarizing, a cost comparison is outlined in Table II.

TABLE II

CLOSED CYCLE & THREE-PIPE SYSTEM:

Increases cost due to:	*Decreases cost due to:*	*Net effect is:*
1 double condenser	Combined heating and cooling piping. Eliminates duplicate heating and cooling coils. Eliminates separate heating and cooling controls. Eliminates double fans. Eliminates triple dampers. Eliminates 100% O. A. ducts. Eliminates 100% O. A. exhaust.	3 small pipes replace 4 large pipes. 1 coil for heating & cooling. 1 control for heating and cooling. Substantial space savings. Constant ventilation. Improved filtering. Lessens freezeup hazard.

INCREASED AIR COIL SURFACE:

Increases cost of:	*Decreases cost of:*	*Net effect is:*
air coils	Circulating pipes, insulation, control valves. Circulating pumps. Compressors (if staged). Supplemental heat plant by use of closed cycle.	Wide water range, less water circulated. Reduced pump horsepower. Reduced compressor horsepower.

STAGED CYCLE & USING MORE STAGES:

Increases cost due to:	*Decreases cost due to:*	*Net effect is:*
using more compressors	Smaller motors & compressors.	Reduces power cost. Smaller chilled water, piping, control valves, pumps, motors, compressors.

NEXT MONTH

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REVIEWS . . .
of Technical Session Papers

PICTURES . . .
of Technical Program and social events

Cooling rates of apples

packed in different bushel containers and stacked at different spacings in cold storage

D. V. FISHER

Until recent years western apples have been packed exclusively in wooden bushel boxes, with apples paper-wrapped and tightly placed in the container to produce a high crown bulge. The boxes were stacked in cold storage at a space of 3½ in. between rows and the stacks placed bulge to bulge in the rows. This resulted in fairly satisfactory cooling rates because of exposure of 4 out of 6 sides of the box to cooling air. Under these conditions reduction of fruit temperature from field heat to 32 F was normally accomplished in one week.

The present day packing method is to place apples unwrapped in cells or trays in corrugated cardboard bushel containers. These containers are of various lengths depending on the number of apples in them. When placed in cold storage, stacks of containers are spaced at distances varying from one to four in. With narrower spacings, poor row alignment often causes blockage of air flow between adjacent rows of stacks. Furthermore, since cardboard containers have no "bulge" they are often placed tightly stack to stack in rows, thus reducing the number of air-free surfaces around the box from six to two. The result has often been that cardboard containers have not cooled satisfactorily.

Following complaints from packing house operators of poor cooling in the 1957 season, a series of cooling tests was carried out using wood boxes with cardboard tray packs and cardboard cell packs with various types of pads and partitions.

Work reported by Sainsbury and Schomer[2] on cooling of pears in wood and cardboard cartons using different stacking arrangements, indicated that tight stacking impeded heat loss, more so with cardboard than wood. The greater number of sides exposed to air movement the faster was the heat loss. Differences between stabilized storage temperatures of pears in cardboard cartons and those in wood boxes were much greater when the containers were tightly stacked than when more of their surface was exposed to air flow. Large differences were found in rate of fruit cooling between different storage plants, probably attributable to degree of efficiency of air circulation within the plants.

Further data by Sainsbury[1] on the use of vents in cardboard cartons to improve cooling of apples indicated that apples in unvented cartons often took three times longer to cool than those packed in wooden boxes, and that the fruit remained on the average 2 F warmer than in wooden boxes.

Cartons with large vents plus perforated trays cooled in about the same time as fruit in standard wooden packs. Smaller vents and absence of vents increased cooling time. With a stacking pattern exposing both ends of the cartons, the best venting resulted in a temperature difference between carton and air of 0.8 F and poorest venting, 2.0 F.

EXPERIMENTAL

Cooling of well spaced individual wood and cardboard containers—In a preliminary study of cooling rates of apples in cardboard vs wood containers, the object was to determine rate of heat loss under the most favorable conditions. The apples in these containers were raised to a temperature of 66 F, they were placed in a 32 F room with good air circulation, and the containers were so placed as to have free air circulation on all sides. A remote

Here is a pertinent study of cooling rates of apples under commercial storage conditions. Comparison of apples stored in wood and cardboard cartons shows significant differences in cooling time and in final equilibrium temperatures. Containers stacked solidly on pallets took twice as long to cool to 32 F as similar containers stacked loosely. Also, equilibrium temperature of fruit in solid pallet loads was 0.5 F higher than in spaced loads.

D. V. Fisher is Head, Pomology Section, Canada Department of Agriculture. This paper was presented at the ASHRAE 67th Annual Meeting, Vancouver, B. C., June 13-15, 1960.

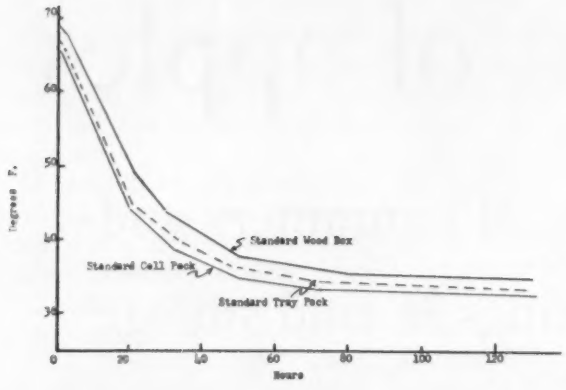

Fig. 1 Cooling time required to reduce temperature of standard cell packs, tray packs and wood boxes from 66 F to equilibrium with full exposure of all surfaces of containers to 32 F air

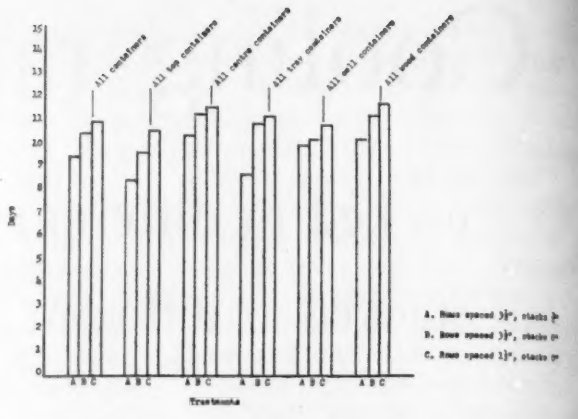

Fig. 2 Comparison of cooling rates of apples from 64 F to equilibrium in an air temperature of 31 F with different containers, stack spacing arrangements and container location in the stack

reading thermocouple was placed in the center apple of each container. A series of graphs was prepared which showed that rapid cooling took place with all types of containers but that in every instance wood boxes were slightly slower in cooling than cardboard boxes.

A typical graph is given in Fig. 1. In each run, temperatures of approximately 33 F were realized in 80 hrs. The slightly faster cooling performance with cardboard might be explained by absence of insulating effect of paper wraps used in wood boxes and by thinner sides, bottoms and ends with cardboard.

Cooling of wood box and cardboard containers using different stacking methods—In the winter of 1958 a comprehensive cooling experiment was set up in the B.C. Fruit Shippers Summerland cold storage plant to compare cooling rates of Rome apples in wooden boxes and in various vented and unvented tray and cell cartons. All containers held 120 or 125 pieces of fruit except for the frame-end cell pack which held 200 apples. All of the fruit was unwrapped except in the standard wood boxes. The following containers were used in the experiment:

(1) Standard wood box.
(2) Friday tray pack carton with full telescope top – no vents.
(3) Friday tray pack carton as under (2) except 2 vents each side and end, 2 x ½ in.
(4) Friday tray pack carton as under (2) except 4 vents each

TABLE I

EFFECT OF CONTAINER CONSTRUCTION AND SPACING ARRANGEMENT ON COOLING OF APPLES AT AIR TEMPERATURE OF 31 F

Container Description		COOLING PERFORMANCE A Rows 3½ in. apart, stacks ¾ in. apart From 64 F to 35 F Days	A From 64 F to equil. Days	A Equil. temp. F	B Rows 3½ in. apart, stacks 0 in. apart, From 64 F to 35 F Days	B From 64 F to equil. Days	B Equil. temp. F	C Rows 1½ in. apart, stacks 0 in. apart From 64 F to 35 F Days	C From 64 F to equil. Days	C Equil. temp. F
Tray pack unvented	Top pack	4.5	8.0	32.0	6.5	10.0	32.6	6.0	11.0	32.8
	Centre pack	5.5	10.0	32.8	8.5	11.5	33.6	7.5	13.5	33.1
Tray pack with 2 side and 2 end vents	Top pack	2.5	8.0	31.1	4.0	10.0	32.3	4.5	11.0	32.5
	Centre pack	4.5	9.0	32.7	8.5	11.5	33.7	4.0	10.0	31.9
Tray pack with 4 side and 4 end vents	Top pack	3.5	7.0	32.3	5.5	11.5	32.1	4.5	10.0	32.3
	Centre pack	4.5	12.0	32.7	8.0	12.5	33.4	5.5	12.5	33.6
Tray pack, frame-end with 2 end vents	Top pack	4.0	7.0	33.5	3.0	9.0	31.7	5.5	10.0	32.9
	Centre pack	3.5	8.0	32.0	4.0	10.0	32.0	7.0	11.5	33.1
Cell pack unvented	Top box	4.0	11.0	32.6	5.0	9.0	32.6	6.0	10.0	33.1
	Centre box	6.0	11.5	33.0	8.0	11.5	34.0	9.5	10.5	34.4
Cell pack with 2 side and 2 end vents	Top pack	2.5	11.0	32.0	3.0	9.0	31.7	3.5	8.0	32.3
	Centre pack	3.5	8.0	32.5	4.5	10.5	32.3	6.0	9.5	33.2
Cell pack with 4 side and 4 end vents	Top pack	3.0	7.0	32.0	7.0	10.0	33.0	3.5	12.5	32.9
	Centre pack	4.5	12.0	32.0	6.5	11.5	32.9	7.0	12.0	32.6
Cell pack, frame-end with 4 end vents	Top pack	3.0	7.0	32.7	4.0	9.0	32.7	6.0	11.0	32.9
	Centre pack	5.5	12.0	32.9	9.0	10.5	34.1	9.5	12.5	33.7
Standard wood box	Top pack	4.5	9.0	32.8	4.0	10.0	32.3	6.0	11.0	33.7
	Centre pack	4.5	11.5	32.2	4.5	12.5	32.0	6.5	12.5	32.6

TABLE II

STATISTICAL SIGNIFICANCE OF DIFFERENCES IN COOLING RATES OF DIFFERENT PACKAGES

Packages	Means, cooling time from 64 to 35 F Days	Means, cooling time from 64 F to equilibrium Days	Mean temp. at equilibrium F
Tray pack, unvented	6.42	10.67	32.82
Tray pack with 2 side and 2 end vents	4.67	9.92	32.37
Tray pack with 4 side and 4 end vents	5.25	10.92	32.73
Tray pack, frame-end, with 2 end vents	4.50	9.25	32.53
Cell pack, unvented	6.42	10.58	33.28
Cell pack with 2 side and 2 end vents	3.83	9.33	32.33
Cell pack with 4 side and 4 end vents	5.25	10.83	32.57
Cell pack, frame-end, with 4 end vents	6.17	10.33	33.17
Standard wood box	5.00	11.08	32.60
L.S.D.'s			
P = .05	1.09	1.14	.54
P = .01	1.50	1.57	.75

side and end, 2 x ½ in.

(5) Friday tray, frame-end pack carton as under (2) except 2 vents each end 6 x ¼ in. and fold-over stapled top.

(6) Cell pack carton with no vents.

(7) Cell pack carton with 2 vents, each side and end, 2 x ½ in.

(8) Cell pack carton with 4 vents, each side and end, 2 x ½ in.

(9) Cell pack carton, frame-end, 4 vents each end, 2 x ½ in.

The fruit was heated to approximately 64 F and placed in 6-high stacks within regular warehouse rows of fruit of the same package size in the commercial 31 F storage to which reference has been made. The block containing experimental packs was buffered on either side with regular commercial fruit, and buffer stacks were placed between the experimental stacks in the rows to complete the row arrangement and to provide for commercial cooling conditions.

Air circulation was representative of good commercial practice at about 1100 cfm per ton of refrigeration with air delivery parallel to row alignment. A thermocouple was inserted into the apple nearest the center of the package in each top and middle box of each experimental stack of fruit. Using remote leads resistance readings were taken daily until an equilibrium temperature had been achieved.

Three separate cooling runs were performed, in each case with the same fruit and thermocouples. After completion of a run the containers were removed from storage, warmed to a temperature of 64 F and then replaced in storage for the succeeding tests. The three tests reported on represent:

(A) Stacks ¾ in. apart, rows 3½ in. apart.

(B) No space between stacks, rows 3½ in. apart.

(C) No space between stacks, rows 1½ in. apart.

The data from this experiment are presented in summarized form in Table I and Fig. 2 showing cooling time (to the nearest half day) from 64 to 35 F and from 64 F to the final equilibrium temperature attained by each package.

These data have been further statistically examined (Tables II, III, IV) showing means for all packages, for locations and for spacings with appropriate L.S.D.'s.

Packages – Table II indicates that unvented tray and cell packs took 6.42 days to cool from 64 to 35 F compared with 5.00 days for the standard wood box and close to this time for most other containers. The differences in cooling rates between unvented tray and wood containers were highly significant (P = less than 0.01).

Those between unvented cells and wood containers were significant (P = less than 0.05). The cell pack frame-end, vented package was also significantly slower in cooling than the wood container but this is probably explained by it being the only container holding 200 apples, all other packages holding 120 or 125 apples.

The additional number of cells

TABLE III

STATISTICAL SIGNIFICANCE OF DIFFERENCES IN COOLING RATES OF TOP AND CENTER CONTAINERS IN STACKS

Location	Means, cooling time from 64 to 35 F Days	Means, cooling time from 64 F to equilibrium Days	Mean temp. at equilibrium F
Top container	4.41	9.52	32.50
Centre container	6.15	11.13	32.93
L.S.D.'s			
P = .01	.71	.75	.36

TABLE IV

STATISTICAL SIGNIFICANCE OF DIFFERENCES IN COOLING RATES OF PACKAGES SUBJECTED TO DIFFERENT SPACING ARRANGEMENTS

Spacing	Means, cooling time from 64 to 35 F Days	Means, cooling time from 64 F to equilibrium Days	Mean temp. at equilibrium F
A. Rows 3½ in. apart stacks ¾ in. apart ...	4.08	9.39	32.43
B. Rows 3½ in. apart, stacks 0 in. apart	5.75	10.53	32.72
C. Rows 1½ in. apart, stacks 0 in. apart	6.00	11.06	32.98
L.S.D.'s			
P = .05	.63	.66	.31
P = .01	.87	.89	.42

TABLE V

COOLING RATES OF APPLES PACKED IN CARDBOARD CARTONS AND PLACED WITH DIFFERENT SPACINGS ON PALLETS SET THREE HIGH

Container and type of stacking	Thermistor location	Time to Cool from 64 F to: 35 F Days	Equilibrium Days	Equilibrium temperature F
Frame-end tray carton, 8 cartons per pallet layer, no air spaces	Top pallet, centre	10.0	15.0	32.4
	Top pallet, outside	7.0	15.0	32.3
	Bottom pallet, centre	9.5	14.5	33.4
	Bottom pallet, outside	8.0	14.0	33.4
Frame-end tray carton, 7 cartons per pallet layer, lock-stacked to provide air spaces	Top pallet, centre	4.0	7.0	31.5
	Top pallet, outside	4.0	7.0	31.1
	Bottom pallet, centre	4.5	8.0	32.3
	Bottom pallet, outside	4.5	8.0	32.8
Tray carton, 7 cartons per pallet layer, lock-stacked to provide air spaces	Top pallet, centre	4.5	9.5	31.5
	Top pallet, outside	3.5	6.5	31.2
	Bottom pallet, centre	3.5	7.0	31.2
	Bottom pallet, outside	4.0	8.6	32.0

would result in a slower rate of heat release. On the other hand, the cell pack with 2 side and 2 end vents cooled significantly faster from 64 to 35 F than the standard wood box (3.83 vs 5.00 days).

The period from start of cooling to attainment of equilibrium temperature ranged on the average from 9.25 to 11.08 days for different packages. The tray pack with 2 side and 2 end vents, the frame-end tray pack with 2 end vents, and the cell pack with 2 side and 2 end vents cooled slightly but significantly faster than the standard wood box.

The only significant difference found between final equilibrium temperature of the standard wood box (32.60 F) and other packages was that of the frame-end cell pack with 4 end vents (33.17 F). This was probably the result of greater insulation from having 200 fruit cells in the latter compared with only 125 apples in the former packages. Where packages with equal fruit counts were compared, cooling performance was almost as good in one package as the other, and final temperatures similar.

Box location in stack – Table III provides a comparison between rate of cooling of apples in top containers and those in center containers of stacks. The cooling time averaged 1.5 days longer in center than in top stacks and the equilibrium temperature was 0.43 F higher. These differences were both highly significant.

Stack and container spacing—Data are presented in Table IV to show the effect of spacing on package cooling performance. Wide (A) spacing gave a highly significant improvement in rate of cooling and in final fruit temperature compared with narrow (B and C) spacing. The mean time required for all packages to drop in temperature from 64 to 35 F was 4.08, 5.75 and 6.00 days respectively for A, B and C spacing.

To reach final equilibrium temperatures of 32.43, 32.72 and 32.98 F, required 8.3, 9.7 and 10.5 days respectively for A, B and C spacing. The higher final temperatures associated with close spacing are probably of much more practical importance than the slightly slower cooling rates.

Cooling of cartons of apples stacked in pallets in commercial storages – Information on cooling rates of apples, 120 to the carton, packed in cardboard cartons and stacked on pallets in a commercial cold storage is presented herewith. The test included:

(1) Fruit in frame-end tray cartons, placed solidly, 8 to a layer, 5 layers high on 43 x 49 in. pallets, stacked 3 high.

(2) Fruit in frame-end tray cartons, placed 7 to a layer for ventilation, 5 layers high, on 43 x 49 in. pallets, stacked 3 high.

(3) Fruit in ordinary tray cartons, lock-stacked to provide air spaces between cartons for cooling and placed 7 to a layer.

Thermocouples were placed in the center of an outside and inside carton, both on the top pallet and the bottom pallet, with similar pallet stacks surrounding the test stack on all sides. Air temperature in the room was 30.5 F and varied little between top and bottom pallets because of good circulation from an effective forced air circulation system.

Cooling times reported are for reducing fruit temperature from 64 to 35 F and to final equilibrium temperature. The results are summarized in Table V. It is apparent that cooling time, from 64 F to either 35 F or equilibrium, was twice as long with tight as with loose spacing, and equilibrium temperatures with solid loading were 1 F higher than with spaced loading. With both types of spacing, top pallet loads averaged 1 F lower in temperature than bottom pallet loads.

The cooling performance of regular cardboard tray cartons on pallets spaced 7 to a layer was similar to that of the frame-end cartons similarly spaced. The advantage of frame-end cartons is their resistance to crushing when pallet is loaded; and the main disadvantage, instability of pallet loads when lock stacked.

DISCUSSION

Studies on cooling rates of containers freely exposed on all sides to good air movement pointed to the fact that with good surface exposure quite rapid cooling took place (66 to 33 F in 80 hr). This was confirmed in actual warehouse tests where various containers were stacked 6 high at different spacings in a commercial cold storage plant. Cooling tests in another storage using fruit loaded on pallets stacked solidly and spaced on pallets (lock stacks) demonstrated that spacing of containers reduced cooling time by 50 per cent, and lowered final holding temperature by about 1 F.

Cartons with vents of various sizes and numbers did not differ significantly from wood boxes in rate of cooling. This is probably explained by the blockage of ventilation within containers by the presence of cells, trays and fruit.

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Engineers and World Trade

GEORGE F. TAUBENECK
Member ASHRAE

Author Taubeneck is no stranger to other members of ASHRAE. He records this article as being based on his 107th talk before ASHRAE, or predecessor ASHAE and ASRE chapters or sections, in the course of 30 years.

Are we pricing our goods out of foreign markets? Are we importing unemployment by establishing branch factories abroad? Do we need protective tariffs to combat low-paid labor in foreign countries? Or can we re-establish ourselves as purveyors of quality goods to the world?

Answers to the first three questions should be negative. Response to the last query, if ASHRAE engineers are alerted and consulted, can be a resounding "YES."

But isn't this really a job for marketing specialists, rather than engineers? On the contrary, engineers hold the key to a coming "Made in U.S.A." export boom.

In many areas of international commerce—even in our own backyard – foreign manufacturers are beating us on price. They are producing equivalent products (in terms of performance, quality, and utility) with a built-in cost advantage of lower wages. Key word here is "equivalent." In other words, they copy us.

At that point American engineers enter the picture. It is their responsibility—even obligation—to design BETTER, more interesting, attractive, and saleable devices than their foreign competitors can ship over here. That's the best way we can lick the lower-wage bogey.

So far our competitors in Europe and Japan merely have matched several species of our consumer goods. We lead . . . they follow . . . and eventually catch up. Trouble is, we haven't been doing enough leading lately.

Innovation will put us on top again. If and when our engineers develop different products, advanced designs, more attractive styling, we can regain dominance in those world markets we once had all to ourselves.

And don't say we can't do it. Despite all the propaganda you may have read about Russia training more slide-rule artists currently than the U.S.A., we still have many thousands more trained engineers at work than they do. Or any other foreign competitor, for that matter.

Moreover (and this is ultra-important) our political and economic "climate" of individual freedom is much more conducive to invention and innovation than that of the Soviet Union.

Another thing: our emphasis always has been on consumer goods, in contrast to Communist concentration on materials-for-warfare-and-propaganda. (Sputniks and moon-shots comprise an effective method of propaganda warfare.)

Individual freedom to invent, plus a tradition of consumer goods for everyone, are tremendous American advantages . . . if we use them wisely.

We have a handicap, though. It lies in the current Trend Toward Conformity, which tends to affect engineers almost as stultifyingly as it does the Gray Flannel Suit boys who concoct the crime-cowboy-and-corn TV shows.

What America needs is encouragement of enterprising engineers who dare to be different. Our Edisons and Ketterings didn't punch timeclocks or listen to "motivation researchers." They were individualistic laws-unto-themselves. Nor were they slaves to "mass production thinking," re., produce uniformly only those items which vast masses bought yesterday.

By ignoring the wants and tastes of minorities, we opened our markets to European autos and Japanese cameras, among other things. Our attitude has been, as Henry Ford put it so succinctly 40 years ago: "I'll give 'em any old color they want so long as it is black."

Right there is the chief complaint of foreign importers. We design our products for the Ameri-

George F. Taubeneck, Editor & Publisher of Air Conditioning, Heating & Refrigeration News, most recently addressed the ASHRAE Detroit Chapter on April 18. He has prepared this material from his extemporaneous remarks at the request of the Editor.

can "mass market" (which no longer exists, incidentally) and refuse to bother with practical mutations adaptable to local conditions in, say, Zanzibar or Teheran.

A prime example is 50-cycle electricity, which is far more prevalent abroad than our own 60-cycle "juice." Too many of our manufacturers tend to ignore this fact. Result: unhappy customers, unsatisfactory products, market inroads by foreign manufacturers.

Engineers who are willing to examine these local conditions and tastes, and thereafter provide alternate designs to fit odd voltages, wattages, climatic conditions, etc., can help the U.S.A. compete successfully in growingly profitable world marts.

Global philosophers have sensed that American scientists and engineers in toto are suffering a professional guilt complex which has thwarted their normal urges for inventing and pioneering.

What is the root of this massive professional psychosis? The atom bomb. American scientists and engineers perpetrated it, and subconsciously (or consciously) they are unhappy because they can't control it. Here's how all engineers can rid themselves of this unconscious "guilt complex." Aid and abet world trade! The more the latter burgeons, the more probable world peace becomes. Nobody wants to shoot good customers.

"World peace through world trade" could be the theme of America's current drive to step up exports. Here are some of the facets of that program:

1 – For the first time in its history the U.S.A. will guarantee short-term export credit (used chiefly for consumer goods) against loss from foreign governmental action – such as currency exchange regulations, expropriation, acts of war, etc. A small insurance fee will be charged for this protection.

2 – Exhibits at international trade fairs will be stepped up, and the "hard sell" will be substituted for the "don't brag" approach previously in vogue in our representation at these shows.

3 – U. S. embassies abroad will attempt to recruit commercial specialists from among American businessmen, so as to gain on-the-spot salesmen with "know-how" who can move our goods.

4 – Department of Commerce officials also will turn this "hard sell" approach inward. They'll work overtime to convince American businessmen that the latter do have excellent opportunities for gaining extra profits by engaging in world trade. The Small Business Administration will get into this act, too.

5 – Finally, "trade missions" (small groups of U. S. businessmen who go abroad to work out mutual commercial exchange deals, promote exports, etc.) will be multiplied.

There's more to this world trade promotional effort than would appear to the naked eye. Value of the dollar is at stake. Although our exports continue to exceed imports, and our earnings from factory branches and licensees abroad are leaping at an astounding rate, we are losing gold.

Reason: Our gigantic foreign aid donations, plus immense costs of our military establishments on foreign soil, along with the tremendous rise in tourism, have resulted in a total outflow of dollars exceeding our current intake.

Economists call this phenomenon "a balance of payments deficit." If it goes on much longer we'll be in serious trouble at home as well as abroad. That's why, all of a sudden, it has become patriotic to be an exporter, a world trader.

It should be footnoted that not a few American industries will oppose this program. They're the ones who already are hurt by foreign competition. Included in this list are textiles, copper and steel, cameras, electronic products, even automobiles.

Latter industries generally shudder at the bogey of lower-wage rates abroad. That's a factor, of course, but it can be overcome by superior American technological efficiency, and advanced designing.

Our refrigeration, heating, air conditioning, and home automation related industries provide a striking case in point. The good ol' U.S.A. is a world leader in terms of "know how" and advanced designing in these areas.

Almost every refrigeration, air conditioning, heating, and home automation product bought abroad is American in origin. Whether exported, manufactured under license or by a branch factory, it is ours.

Interestingly enough, shipments of U.S.A. products to international markets have been ascending this year. Good reasons why:

Quota-type import restrictions against American goods have been lifted in many European and Near East nations. Also in booming Australia and Japan. The Pacific Ocean area markets alone can absorb (because they desire and can afford) far more of our exports than we've ever shipped them heretofore. And now they've opened the gates.

Filing cabinets in the U. S. Department of Commerce already contain reams of credit reports and other useful information on more than 500,000 foreign establishments which are ready to do business with American firms.

Overseas commercial attaches send back regular economic reports to Washington. From these reports pamphlets are compiled on foreign-trade leads, business conditions, etc. – which will help you get started.

At present your Commerce Department is preparing a card index of foreign concerns which now act, or would like to serve, as agents abroad for American exporters. Already this list totals more than 100,000 cards.

All this governmental activity is praiseworthy. However, an export "push" won't get off the ground unless engineers concern themselves with the problems of redesigning our products to suit foreign markets. No longer is our design target "a little old lady in Kansas City."

Nowadays we'll need to satisfy Karl Rumpelsmayer in Vienna, Henk de Groot in Amsterdam, Pierre Bardot in Paris, Antony Something-Smith in London, Charlie Chan in Hong Kong, Mbwawah Ngongongg in Tanganyika, and Mathilda Waltzing in Melbourne.

Actually, this situation should breathe new life into the engineering profession. It will be stimulating—even exciting—to solve the new and variegated problems involved with world trade.

Secretary Emeritus

A. V. Hutchinson

1896-1960

Executive Secretary of the American Society of Heating, Refrigerating and Air Conditioning Engineers from its merger-formed start in 1958 until his retirement in 1959 and similarly allied with precedent American Society of Heating and Air Conditioning Engineers for 36 years, Aubry Vaughan Hutchinson, Secretary Emeritus, died in Lafayette, Ind., June 6, 1960.

Mr. Hutchinson joined the American Society of Ventilating Engineers (as ASHAE was then termed) in 1922 as Manager of Publications. He was appointed Secretary in 1926 and Executive Secretary in 1950. When ASHVE became ASHAE, he continued as Executive Secretary and Manager of Publications.

He preceded his 37-year career with the Society by entering the publishing field with David Williams Company, New York, in 1914-15. He then served from 1916-17 as assistant and later associate editor of Metal Worker, Plumber and Steam Fitter in New York and as western editor for Sheet Metal Worker and Heating Engineer in Chicago from 1920-21.

Born November 23, 1896 in Brooklyn, N. Y., Mr. Hutchinson was the son of Robert Lee and Caroline Estelle (Vaughan) Hutchinson. He attended schools in Ridgewood, N. J., taking special courses in journalism and engineering. In 1933 he married Janis Louise Barr, who survives him.

Mr. Hutchinson served with the U. S. Army in 1918-19. He was a member of the Society of American Military Engineers and the American Legion. Until his retirement, Mr. Hutchinson made his home for many years in Wyckoff, N. J.

Mr. Hutchinson's interests and activities were broadly allied with the engineering, publishing and industry functions of the Society with which he was so long associated. In his 37 years of service, Mr. Hutchinson witnessed the growth of the Society from a small group to that of one of the five largest professional engineering societies. At the time of his retirement then President Arthur J. Hess paid high tribute to his work, stating, "Probably the most important contribution, among the many he made, was his work on the 'Guide' which he built up into the 'Bible' of the Industry and one of the most respected publications of any technical society. To Mr. Hutchinson we owe much more than we can say in words, or do in deeds."

Meetings ahead

August 23-25—Sixth Cryogenics Engineering Conference, Boulder, Colo.

October 10-12 — American Gas Association, Annual Convention, Atlantic City, N. J.

October 20-23—Refrigeration Service Engineers Society, Annual Convention, Portland, Ore.

October 25-27 — American Standards Association, 11th National Conference on Standards, New York, N.Y.

October 28-November 2 — Air Conditioning and Refrigeration Wholesalers, Silver Anniversary Convention, SS Hanseatic.

October 29-November 4—Oil Heat Institute, Diamond Jubilee Cruise and Management Conference, HMS Queen of Bermuda.

October 31-November 3 — Institute of Boiler and Radiator Manufacturers, Semiannual Meeting, Absecon, N.J.

November 14-17—National Warm Air Heating and Air Conditioning Association, 47th Annual Meeting, Cleveland, Ohio.

November 18-22 — Air Conditioning and Refrigeration Institute, Annual Meeting, Hollywood Beach, Fla.

November 20-23 — Refrigeration and Air Conditioning Contractors Association, Annual Meeting, Miami, Fla.

November 27-December 2—American Society of Mechanical Engineers, Annual Meeting, New York, N. Y.

November 28-December 2—24th National Exposition of Power and Mechanical Engineering, New York, N. Y.

December 1-2 — National Association of Practical Refrigerating Engineers, Annual Meeting, St. Louis, Mo.

February 13-16—American Society of Heating, Refrigerating and Air Conditioning Engineers, Semiannual Meeting, Chicago, Ill.

February 13-16 — 15th International Heating and Air-Conditioning Exposition, Chicago, Ill.

People

H. W. Rainey, Jr., is now Marketing Manager of Sturtevant Div, Westinghouse Electric Corporation. Immediately prior to his new appointment, he was General Sales Manager. An alumnus of Georgia Tech, he joined B. F. Sturtevant Company after graduation. Sturtevant was acquired by Westinghouse in 1945 and, in 1955, Rainey was made Regional Manager. He became General Sales Manager in 1959.

W. E. Robb, in his new capacity as Project Engineer with Webster Engineering Div, Midland-Ross Corporation, will be responsible for design and development of combustion equipment and associated automatic controls. Educated at the University of Kansas, with B. S. degrees in both mechanical and electrical engineering, he was formerly Chief Engineer of Gordon and Piatt, Inc.

Wendell Webster will fill the newly created position of Chief Engineer, Air Conditioning Products, for Coolerator Div, McGraw-Edison Company. He has been associated with Coolerator and its predecssor for more than ten years.

Joseph S. Kearney, President of Northwestern Heating and Plumbing Company, has been elevated to the Presidency of the Mechanical Contractors Association of America from his former position of Senior Vice President. ASHRAE Life Member **Leon L. Munier** was, at the same time, elected Treasurer.

Graham S. McCloy, a Westinghouse Electric Corporation engineer since his graduation from Georgia Tech in 1929, has been named Manager of Engineering for the major appliance div. Appointed Development Engineer in 1931, he was made Manager of the technical engineering sect in the Springfield, Mass., plant in 1949 and was transferred to the Columbus, Ohio, plant last year when the company consolidated its compressor-driven product engineering there.

Milton Levine, appointed District Sales Manager in metropolitan New York and Westchester County for Welbilt Air Conditioning and Heating Corporation, was formerly a sales representative for Cold Wave Cooling Corporation.

W. Walter Timmis, deceased, had served as a mechanical engineer for Purdue University. Formerly owner of his own firm, Walter Timmis & Company, he had been employed by such organizations as Kewanee Boiler, C. A. Dunham, American Radiator Company (as Manager, Air Conditioning and Controls) and Air Temp Company (as President). An ASHAE member since 1925, he served on the Guide Committee from 1937-39 and had, at various times, been Vice President, President and a member of the Board of Governors of New York Chapter.

Walter A. Bornemann, an ASHRAE Life Member, died on May 1st at the age of 67. Educated at Lehigh University and Stevens Institute of Technology, he joined Carrier Corporation in 1919, remaining with that organization until his retirement in 1958. He was a past-President of Valley Forge Chapter of the Pennsylvania Society of Professional Engineers.

C. D. Bassett has merged his consulting engineering practice with Cummins & Barnard, Inc., engineers. He will serve as Vice President and Director.

R. C. Crowe, recently named Sales Manager of the New York District for Wolverine Tube Div, Calumet & Hecla, Inc., has been appointed Manager of Product Sales. Joining Wolverine in 1950, he has served as Industrial and Public Relations Supervisor as well as in various sales capacities.

Gilbert Nelligan is Sales Representative for Young Radiator Company in the St. Petersburg, Fla., area. He has had 23 years of sales engineering and management experience with Fitzgibbons Boiler Company.

George W. Brooke, appointed Manager—Methods and Procedures of American-Standard's Industrial Div, has worked in engineering and material control with Acme Industries and was a systems analyst with American-Standard's Detroit Controls Div.

E. B. Moore, previously Market Liaison Engineer, has been appointed Publicity Director of Vapor Heating Corporation. He joined Vapor's Engineering Dept in 1949 as Director of Research and has held various engineering posts with the company, including Chief Mechanical Development Engineer and Assistant Chief Engineer in the Car Heating Div.

James W. McGuire has been appointed Sales Agent of CRS Industries, Inc., in the metropolitan New York area, central New York State, Connecticut and Rhode Island.

James L. Brown has been named District Sales Manager for Stewart-Warner Corporation's Heating and Air Conditioning Div. He will be responsible for sales activities of dealers in 25 northern Ohio and southern Michigan counties.

Michael H. Pelosi, Jr., has been appointed Director of Engineering for CRS Industries, Inc. Formerly Chief Engineer of United States Air Conditioning Corporation and Bryant Air Conditioning Corporation, he has a background of 14 years in designing, manufacturing and consulting services in the air conditioning, heating and ventilating industries.

T. R. Kearney, recently retired after 25 years with Ansul Chemical Company, has been appointed Vice President, Sales, of Sealed Units, Inc. He was Refrigeration Manager at the time of his retirement from Ansul.

A. E. Kline, Vice President in Charge of Engineering, Coolerator Div, McGraw-Edison Company, died April 29th after a long illness. A pioneer in engineering development of both room air conditioners and dehumidifiers, he had served the div and its predecessor companies for more than ten years.

Edward F. Rau, appointed Sales Engineer in the Metals Div of Olin Mathieson Chemical Corporation, has been assigned to the St. Louis territory. After attending the University of Wisconsin, he joined, in 1936, Hamilton Manufacturing Company and in 1945 became a sales engineer and later Service Manager for Manitowoc Equipment Works. Prior to joining Olin Mathieson he was Chicago Regional Manager for Bohn Aluminum & Brass Corporation.

T. I. Harriman, appointed Manager of Pricing Service, and **R. L. Graham,** named Manager of the Los Angeles office, are ASHRAE members assuming new positions with American Radiator & Standard Sanitary Corporation. Graham is the former Manager of the Moline office.

Jerome Pinkus, appointed to the newly created post of Chief Products Engineer, Heating and Air Conditioning, for Utility Appliance Corporation, has been with the organization for nine years.

Otto J. Nussbaum, Chief Engineer with Kramer-Trenton Company, will start a two-month leave of absence from that organization on July 1st, in order to accept a temporary appointment with the United Nations as Technical Advisor to the Polish Government at Warsaw.

Donald K. Miller, former District Engineer for the southwest region of York Div, Borg-Warner Corporation, in Houston, has been transferred to the main office in York, Pa., as Refrigeration Engineer. A chemical engineering graduate of Missouri University, he was employed by Ralston Purina as a quality control engineer before coming to York in 1951.

Sidney A. Shuman, General Manager of the Fluid Packed Pump Company Div of National Supply Company, has been transferred to Pittsburgh as Assistant to the President. With the div since 1936, when he joined as a sales and service representative, he has been district sales manager in Louisiana, div sales manager in Oklahoma, mid-continent manager and general sales manager. He served as Executive Vice President from 1953 until the purchase of Fluid Packed by National Supply in 1959.

Others are saying——

that . . . . . made from strong paper impregnated with special waterproof resins or lacquers, a new honeycomb insulation material has a thermal conductivity coefficient of 0.033 to 0.045 kcal/m² hr C. To improve thermal insulating properties of the material, the surface of the films forming the air spaces are made heat-reflectant. Honeycomb structure formed by the films, together with the reflecting surfaces, give the material its low degree of thermal conductivity. *Refrigeration Techniques (Xolodil'naya Technika), April-May 1960, p 30 (Russian).*

that . . . . work carried out at the National College for Heating, Ventilating, Refrigeration and Fan Engineering to find a method which uses an electronic computer to size the pipes and pump of an accelerated hot water system has evolved and applied a procedure. Two problems solved by this method involve a simple six-pipe heating system and a system comprised of a network of 76 pipes. First of the two problems was also worked out manually and was in agreement with the computer's solution. *Journal of the Institution of Heating and Ventilating Engineers, April 1960, p 23 (British).*

that . . . . . experiments in which a radioactive gas was used to measure air movements occurring in a detached suburban house showed that excessive air changes resulted from an open fire, but that these were reduced to a more desirable level by using a freestanding convector heater with restricted throat. When rooms were heated by hot water radiators, excessive air changes still took place unless the chimney was sealed or a freestanding convector fire with throat restrictor at max closure was installed. Investigation made into the behavior of a forced-circulation ducted warm air system, with ducting not provided from each room for return air, showed that with doors closed much of the warm air was lost through window areas, even when

(*Continued on page* 80)

Standards can be effective within 60 days of publication

A. T. BOGGS, III
ASHRAE Technical Secretary

At the recent Annual Meeting, an improved method of adopting Society standards became effective by membership vote. The original By-laws for ASHRAE were based upon the previous By-laws of the precedent Societies. Final adoption of a standard previously required membership vote at a national meeting. It was recognized that but a small percentage of members attend national meetings and of this number, an even smaller percentage was likely to be familiar with standards being adopted.

In some instances, this meant that following publication of a proposed standard in the Society JOURNAL, it was necessary to wait six months or more before such standards could be adopted and disseminated. The new By-law makes it possible to put into effect a new standard or a revision of an existing standard within 60 days following publication in the JOURNAL.

This procedure will not deprive the readers of an opportunity to comment on the standard. The Board of Directors will be the final authority and before their approval for adoption, it will be necessary to indicate that all comments on a standard have been justified. The Standards Committee feels that this procedure will enable increased development and promulgation of industry standards. Notice of effective dates of new standards will appear on this page.

ASA: For the first time the U.S. was represented at a meeting of an international technical committee whose objective is the unification of drawing practices throughout the world. The **ISO Technical Committee 10** on Engineering Drawing held a plenary meeting in Lisbon, Portugal in May. The ASA representative who attended this meeting will report to American industry on the necessity of increased participation in this work. The meeting was concerned primarily with reconciling international viewpoints with respect to the indication on drawings of tolerances of form, tolerances of position, and the order of their inscription. American drawing principles have been coordinated to a large extent with those of Canada and Great Britain. In the U.S. the unification of drawing practice has been in progress since 1926 within Committee Y14. The ISO was founded in 1946 to develop common international standards in the fields of technology and engineering. ISO Recommendations are issued for voluntary acceptance by member nations. American interests in ISO are represented by ASA.

ASA: ASA has been requested to call a General Conference on **automation standardization.** This conference has been set tentatively for September 20, 1960 at ASA headquarters. The purpose of the conference is to discuss the implications of the problem of automation and possible solution through ASA procedures. ASA reports that few standards exist that embrace the concepts of this new technology. To prevent the confusion that can result from the development of divergent industry standards is the objective of the conference.

NEMA: The Adsorption Section of Nema has announced availability of an expanded edition of Nema **standards publications for adsorption equipment.** The new edition covers Definitions, Determination of Moisture Vapor Dew Point in Pressure Ranges of 0-10 psig and 10-500 psig, and Procedures for Testing Solid Adsorption-Type Dehumidifiers in the same two pressure ranges. Copies are available from the National Electrical Manufacturers Association, 155 East 44th Street, New York 17, N. Y., at 40c each.

ARI: A second-quarter **Directory of Certified Unitary Air-Conditioners** is available from ARI. The new directory lists some 2150 unitary models of 48 manufacturers. Copies are available through dealers and contractor outlets of the participating manufacturers or Better Business Bureaus. Promotion of the certification program has been authorized by participating companies and as one phase of this ARI in cooperation with NAM has prepared a film for use on TV stations outlining the program and the benefits of air conditioning generally. This TV film is being aired over more than 300 stations and additional prints are available to the industry for showings at local groups. Information concerning this program is available from the Air Conditioning and Refrigeration Institute, 1346 Connecticut Avenue, N.W., Washington, D. C.

ASHRAE STANDARDS PROJECTS

STANDARD No. PROJECT COMMITTEES FOR PROPOSED REVISIONS

16-56R Method of Testing for Rating Room Air Conditioners
Chairman: R. H. Meyerhans
Liaison: W. L. McGrath
Members: C. G. Coyne, K. D. Crook, R. W. Gilmer, E. Gmoser, P. E. Kolb, J. R. Schreiner, E. H. Schwenker, E. C. Tanner

17-48R Methods of Testing for Rating Thermostatic and Constant Pressure Refrigerant Expansion Valves (ASA B60.1-1950)
Chairman: D. C. Albright
Liaison: J. A. Schenk
Members: F. Y. Carter, K. E. Wilson, W. F. Wischmeyer
Alternates: W. Stafford, D. S. Sterner

18-56R Methods of Testing for Rating Self Contained Mechanically-Refrigerated Drinking-Water Coolers
Chairman: W. Taylor
Liaison: F. J. Reed
Members: F. O. Graham, P. R. Lynn, F. J. Reed, E. W. Scott

20-41R Methods of Testing for Rating Remote Air-Cooled and Evaporative Condensers
Chairman: R. H. Merrick
Liaison: A. S. Decker
Members: R. J. Campbell, C. Collins, C. A. Dubberly, T. F. Facius, H. E. Kunkel, R. C. Lower, O. J. Nussbaum, M. A. Ramsey, L. R. Smith, G. Wood

22-48R Method of Testing for Rating Water-Cooled Refrigerant Condensers
Chairman: L. R. Smith
Liaison: A. S. Decker
Members: P. K. Barker, J. G. Bergdoll, C. E. Drake, R. L. Jones, W. O. Kline

24-57R Methods of Testing for Rating Liquid Coolers
Chairman: R. L. Jones
Liaison: A. S. Decker
Secretary: R. A. Mayer
Members: P. K. Barker, J. D. Bergdoll, L. R. Smith

47-35R Method of Testing for Rating Return-line Low-vacuum Heating Pumps
Chairman: W. M. Wallace, II
Members: H. E. Adams, H. S. Bean, A. H. Church, E. G. Keller, L. M. Nechine, A. O. Roche, Jr., L. E. Sterns

53-34R Method of Testing for Rating Steam Unit Ventilators
Chairman: J. W. McElgin
Members: L. A. Cherry, A. F. Hubbard, D. H. Krans, L. G. Miller

PROJECT COMMITTEES FOR PROPOSED STANDARDS

31P Method of Testing for Rating Solenoid Valves for Refrigeration and Air Conditioning Systems
Chairman: J. A. Schenk
Members: D. C. Albright, F. Y. Carter, C. C. Hansen, III, D. S. Sterner, W. F. Wischmeyer
Alternates: J. M. Strauss, K. E. Wilson

36P Methods of Equipment Sound Testing
Chairman: C. M. Ashley
Liaison: S. P. Soling
Consultant: R. J. Wells
Secretary: W. R. Kerka
Members: W. E. Blazier, J. B. Chaddock, J. B. Graham, A. F. Hubbard, D. H. Krans, S. P. Soling

37P Methods of Testing for Rating Unitary Air Conditioning Equipment
Chairman: J. R. Schreiner
Liaison: W. L. McGrath
Members: H. T. Gilkey, G. Keane (AMCA), R. C. Lower, W. H. Mullin, W. A. Spofford, W. E. Ural

39P Methods of Testing for Rating Unitary Heat Pumps for Air Conditioning
Chairman: G. L. Biehn
Liaison: W. L. McGrath
Secretary: R. N. Mahendra
Members: C. H. Belt, H. W. Jobes, R. C. Lower, R. G. McCready, W. A. Spofford

40P Methods of Testing for Rating Heat-operated Unitary Air Conditioners
Chairman: J. G. Reid, Jr.
Liaison: W. L. McGrath
Members: D. E. Heath, R. H. Merrick, B. A. Phillips, F. J. Reed

41P Standard Measurements
Chairman: C. W. Phillips
Liaison: F. J. Reed
Members: F. J. Reed, W. A. Spofford, G. Thompson

49P Minimum Outdoor Air Requirements for Air Conditioning
Chairman: A. Giannini
Members: C. B. Gamble, R. A. Gonzalez, R. J. Salinger, H. G. Strong

51P Testing Air Moving Devices (AMCA Standard)
Chairman: S. Konzo
Members: L. G. Seigel, G. L. Tuve, D. D. Wile

52P Standard for Evaluation of Air Cleaning Devices for General Ventilation and Air Conditioning
Chairman: P. R. Achenbach
Members: W. C. L. Hemeon, R. F. Logsdon, P. E. McNall

54P Method of Testing for Rating Gravity Roof Ventilators
Chairman: R. G. Nevins
Members: G. C. Breidert, F. N. Calhoon, R. B. Engdahl, W. V. Hukill, J. P. Johnson, K. E. Robinson, F. B. Rowley, L. J. Shumaker

Candidates for ASHRAE Membership

Following is a list of 125 candidates for membership or advancement in membership grade. Members are requested to assume their full share of responsibility in the acceptance of these candidates for membership by advising the Executive Secretary on or before July 31, 1960 of any whose eligibility for membership is questioned. Unless such objection is made these candidates will be voted by the Board of Directors.

Note: * Advancement † Reinstatement

REGION I

Connecticut

ENDLER, J. F., JR., Tech. Supvsr., American Brass Co., French Small Tube Div., Waterbury.

JANAZZO, MAURICE, Owner, Wesson Heating & Air Conditioning Corp., Milldale.

SANTA, S. G., Vice-Pres., Santa Fuel Inc., Bridgeport.

Massachusetts

WHITEHOUSE, J. H., Mech. Engr., Ganteaume & McMullens, Boston.

New Hampshire

SCHIAVO, R. D., Field Sales, Engr. Field, Heat Inc., Nashua.

New Jersey

BLOOM, G. M., Appl. Engr., Worthington Corp., East Orange.

BRILEY, H. D.,† Vice-Pres., Thermec Co., Inc., Woodbury.

THAILER, MICHAEL, Design Engr., Worthington Corp., East Orange.

VISCIDO, D. A., Operating Engr., Fireman's Insurance Co., Newark.

New York

BARNES, A. R., Prod. Engr., Carrier Corp., Syracuse.

BONNEY, W. D., Field Engr., American-Standard, Industrial Div., Syracuse.

CAMBELL, R. F., Proj. Engr., Air Contracting Co., Inc., Dewitt.

CLUKEY, J. W., Lab. Tech., Carrier Corp., Syracuse.

DAHLIN, C. R., Drafting Supvsr., Carrier Corp., Syracuse.

FALL, C. W., Prod. Engr., Carrier Corp., Syracuse.

GROSECLOSE, C. E., JR., Chief Engr., in charge of Design, Air Contracting Co., Dewitt.

KRASNOFF, K. J., Mech. Engr., Board of Higher Education, City of N. Y.

LEBLANC, J. J., Prod. Engr., Carrier Corp., Syracuse.

LOGA, F. G., Service Mgr., Air Contracting Co. Inc., Dewitt.

MASS, M. A., Gen. Partner, Cosentini Associates, New York.

OFFERMANN, W. H., Sales Engr., Armstrong Cork Co., Industrial Insulation, New York.

ORBESEN, G. B., Appl. Engr., Carrier Corp., Syracuse.

REISS, A. E.,† Sr. Engr., Carrier Corp., Syracuse.

ROHDE, M. A.,* Supvsr., Gas Htg. Dept., Rochester Gas & Elec., Rochester.

WONDRACK, R. J., Appl. Engr., Aerofin Corp., Syracuse.

WOODCOCK, R. L., Proj. Engr., Air Contracting Co. Inc., DeWitt.

YLITALO, J. E., Lab. Supvsr., Carrier Corp., Syracuse.

Rhode Island

AUSTIN, R. W., Partner, Thoman & Austin, Cranston.

THOMAN, W. C., Partner, Thoman & Austin, Cranston.

REGION II

Canada

BEATON, ALLISTER, Owner, Dunview Sheet Metal Co., Toronto.

CLEROUX, RENE, Owner, Rene Cleroux Plbg. & Htg., Ottawa, Ont.

CUMMINGS, D. S., Architect & Engr., Burgers McLean & Mac Phadyen, Ottawa, Ont.

EDELSON, ELIHU, Owner, Sheet Metal Fabrications, Ottawa, Ont.

FERGUSON, D. G., Repr., Welch & Johnston, Ltd., Ottawa, Ont.

GRIVEL, D. W., Tech. Service Mgr., Fiberglass Canada Ltd., Hamilton, Ont.

HENNING, W. H., Supvsg. Estimator, Rideau Aluminum & Steels, Ltd., Ottawa. Ont.

MACKENZIE, H. A., Owner, Harold A. MacKenzie Assocs., Ottawa, Ont.

MCCARTNEY, M. A., Mgr., Htg. Design Div., EMCO Ltd., Hamilton, Ont.

MCLEOD, D. M., Pres., McLeod Engineering Ltd., Stoney Creek, Ont.

MORRIS, J. R., Mgr., Canadian Comstock Co., Ltd., Ottawa, Ont.

NYMARK, H. C., Asst. Gen. Mgr., Welch & Johnston Ltd., Ottawa, Ont.

SEMENIC, I. A., Proj. Engr., Jas. Keith & Assocs., Montreal, Que.

WALT, P. A., Mgr., Piping Sect. Constr. Div., Provincial Engineering Ltd., Niagara Falls, Ont.

WILLIAMS, R. B., Sales Repr., Keeprite Products Ltd., Town of Mount Royal, Que.

REGION III

Pennsylvania

CONROY, D. J., Mgr. Inds. Sales Div., Electro-Air Cleaner Co., Inc., McKees Rocks.

HUEBNER, G. N., Proj. Engr., Thermecon, Inc., Philadelphia.

KAUFFMAN, N. P., Engr., L. R. Kauffman, Dornsife.

Virginia

BOARD, L. C., Chief Engr., Recony Div., Vinco Corp., Richmond.

REGION IV

Florida

BELINE, M. B.,† Mech. Engr., H. J. Ross Assocs., Miami.

MARTIN, A. R., JR.,* Br. Com. Sales Mgr., Minneapolis-Honeywell Regulator Co., Miami.

MONTGOMERY, J. H., Repr., H. L. McMurry Co., Jacksonville.

OLEN, STEPHEN,† Sales Engr., Minneapolis-Honeywell Regulator Co., Jacksonville.

WENTZ, C. M.,* Design Engr., Florida Power Corp., St. Petersburg.

Georgia

HARRELL, J. A., Htg., Vtg., & A-C. Design Engr., Eastern Engineering Co., Atlanta.

METZ, C. R., Sales Repr., Rubatex Div., Great American Industries, Atlanta.

THORSEN, W. M., Dist. Mgr., A-C. Equip. Sales, Yuba Consolidated Industries, Inc., Atlanta.

North Carolina

ANGEL, I. G., General Partner, Benner & Fields, Inc., Greensboro.

REGION V

Ohio

BODLEY, KENNETH, Sales Supvsr., Permolo Co., Medina.

FAFLICK, W. W., Repr., Cleveland Electric Illuminating Co., Cleveland.

FRODGE, D. B., Mgr. Sales Engrg., Ellis & Watts Products, Inc., Cincinnati.

REESE, D. G., Sales Engr., Johnson Service Co., Cleveland.

SELTMANN, GUNTER, ASHRAE Research Laboratory, Cleveland.

TREAT, J. J.,* Dist. Sales Mgr., Clarage Fan Co., Toledo.

THOMPSON, R. G., Estimating Engr., Dunbar Bros. Inc., Toledo.

Indiana

BOSWORTH, R. W., Sales Engr., Minneapolis-Honeywell Regulator Co., Evansville.

DAVIS, H. C., SR.,* Owner, Elkhart Heating & Refrigerating Co., Elkhart.

FOGGY, G. R., Prod. Engr., Whirlpool Corp., Evansville.
LAMP, J. K., JR., Engr., Whirlpool Corp., Evansville.
LAUGHLIN, J. P., Prod. Engr., Whirlpool Corp., Evansville.
MELCHIORS, G. A., Prod. Service Supvsr., Whirlpool Corp., Evansville.
SCHLENSKER, A. O., Prod. Tester, Whirlpool Corp., Evansville.
SWINGLER, W. F., Sales Engr., Minneapolis-Honeywell Regulator Co., Evansville.

REGION VI

Illinois

MILLER, W. H.,* Sr. Engr., Abbott Laboratories, North Chicago.
OOSTEN, L. S., Vice-Pres., Engrg., Bell & Gossett, Morton Grove.
PATTERSON, M. N., Research Assistant, National Warm Air, Urbana.

Wisconsin

DIESTLER, W. L., Engr., C. A. Hopper Co., Madison.

Michigan

SHUSTER, J. C., Vice-Pres., Leggette-Michaels Co., Grand Rapids.

Minnesota

HINK, LARRY, Sales Engr., Johnson Service Co., Minneapolis.
THORNER, C. A., Sales Mgr., Conditioned-Air Equipment Co., Minneapolis.

REGION VII

Missouri

BEACHAM, C. O., Dist. Chief Engr., Carrier Corp., St. Louis.
ENGEL, N. L., Chief Engr., Monarch Metal Weatherstrip Corp., St. Louis.

Tennessee

BOLING, S. R., JR., Plant Maint. Engr., Dept. of Mental Health, State of Tennessee, Greenville.
EASLEY, W. C.,† Proj. Engr., Velsicol Chemical Corp., Memphis.

REGION VIII

Oklahoma

CONNORS, J. P., Section Chief, Htg., Vtg., & A-C., Western Electric Co., Inc., Oklahoma City.
KING, TED, Owner, Ted King Co., Oklahoma City.

Louisiana

JARVIS, W. L., Vice-Pres., McDaniel Air Conditioning Inc., Shreveport.
JOHNSON, R. L., JR.,* Partner, Johnson-Jones Engineering Co., Shreveport.
SMITH, B. G., Residential Sales, Southwestern Electric Power, Shreveport.
SMITH, W. F., JR., Sales Engr., J. P. Ashcraft Co., Shreveport.

Texas

HOWELL, W. C., Mech. Engr., Kelly Air Force Base, Civil Engrg. Div., 2851st Air Base Wing, San Antonio.
MURPHY, C. W.,† Mgr., Thermal Supply Inc., Austin.
SHURTLEFF, O. O.,† Chief Engr., Chatleff Valve & Mfg. Co., Austin.

REGION IX

Colorado

HUNT, J. M., Vice-Pres., E. N. Murray Co., Denver.

New Mexico

BLACKWELL, W. E., Mgr., Boyd Engineering Co., Inc., Albuquerque.

Utah

PHIPPS, W. B.,† Designer, Lennox Industries, Inc., Salt Lake City.

REGION X

Arizona

PANTANO, M. A., Mech. Engr., Luke Air Force Base, Phoenix.

Canada

BISHOP, D. L., Asst. Bldg. Engr., British Columbia Tel. Co., Vancouver, B. C.
JAMES, F. E., Mgr., Inds. Prods. Div., Flintkote Co. of Canada Ltd., Vancouver, B. C.
WALL, R. E., Sales Engr., Honeywell Controls, Ltd., Vancouver, B. C.

California

AUGSPURGER, F. I., Sales Engr., York Corp., Los Angeles.
DAKOS, D. P., Sales Engr., Koppers Co., Los Angeles.
FARRIS, F. P.,* Sales Engr., Powers Reg. Co., San Francisco.
GERBER, R. L., Design Engr., Oceanside Htg. & Vtg. Co., Oceanside.
GILMORE, D. C., Owner, Mechanical Associates, Anaheim.
YOSHPE, DANIEL,* Cons. Engr., Sacramento.

WHO'S WHO IN ASHRAE

Insofar as possible, these listings will each appear twice a year

ASHRAE OFFICERS, COMMITTEES
See page 62, June JOURNAL

REGION AND CHAPTER OFFICERS
See page 90, May JOURNAL

RESEARCH AND TECHNICAL COMMITTEES
See page 74, February JOURNAL

STANDARDS PROJECTS
See page 63, this issue

INTER SOCIETY COMMITTEES
See page 66, February JOURNAL

Oregon

FUNATAKE, TAI,* Supervision Design Engr., W. Bruce Morrison, Portland.

Washington

DUDEK, W. F., Office Mgr., Trane Co., Seattle.
FELDSTED, R. J., Owner, Feldsted Co., Seattle.
MILLER, H. W., Design Engr. Assoc., Frank W. Jenkinson & Assocs., Spokane.
NIGHTINGALE, W. R., Owner, Air Filter Sales & Service, Seattle.

FOREIGN

Africa

LANCASTER, R. E., Mgr. Engr., Mandilas & Karaberis, Ltd., Freetown, Sierra Leone.

Australia

BURKE, G. C.,* Design & Prod. Engr., Harris Hutchinson Pty. Ltd., Auburn/NSW.
CHAMBERLAIN, K. A., Managing Dir., Sandilux p/c/ Heating & Vtg., Kensington, Victoria.
SHERIFF, W. D., Works Manager, Sandilux Pty. Ltd., Kensington, Victoria.

Ecuador

ROSALES, ALBERTO, Engr., Hydroelectric Projects, Empressa Tecnica Comercial C.A., Quito.

Egypt

MAITY, G. B., Htg. & Vtg. Engr., George Wimpey & Co. Ltd., London.

Hong Kong

LO, H. B.,† Director & Engr., Yee On Hon Co. Ltd., Hong Kong.

India

DEY, S. K., Chem. Engr., Directorate of Industries, Govt. of West Bengal, Calcutta, West Bengal.

Italy

BERGAMINI, UGO, Pres. & Gen. Partner, Soc, Impianti Ing. Bergamini & Co., Rome.
COLLE, AUGUSTO, Appl. Engr., Ideal Standard S.P.A., Milan.
STANZANI, FRANCO, Owner, Franco Stanzani, Rome.

Portugal

SANTOS, E. A., Designer & Appl. Engr., Metalurgica De Longra Ltd., Lisboa.

Singapore

HERVIC, M. P., SR., Sales Engr., International Air Cond. Corp., Singapore.

Trinidad

DUNCAN, A. J., Mgr. Ind. Div., George F. Huggins & Co. Ltd., Port of Spain.

STUDENTS

MISRA, ARUN, Indian Inst. of Technology, Karagpur, W B, India.

What ASHRAE Regions and Chapters are doing

Codes, standards and safety testing were the concern of several Chapters, as members heard addresses on work of the ASHRAE Research Laboratory, Air Moving and Conditioning Association and Underwriters' Laboratories.

WESTERN MICHIGAN . . . In his talk at the May 11th meeting on fundamentals of corrosion, Theodore Davenport, Chemical Consultant for Worthington Corporation, indicated that metal is destroyed by three actions: corrosion, either chemical or electro-chemical, which was defined as the deterioration of metal by reaction with its environment; erosion, or physical destruction by a liquid; and abrasion, physical destruction by a solid. Various factors involved in both types of corrosion were discussed, with stress placed on combinations of materials which could tend to increase the rate of corrosion and methods of controlling this.

NEW YORK . . . Long essential as a fuel for heating, gas has in recent years reached a position of significance as a prime mover in air conditioning and refrigeration systems. Speaking at the May meeting, Harold S. Birkett of Brooklyn Union Gas Company presented a sound slide film on this subject, "Why I specified gas air conditioning." Prepared by the American Gas Association, the film showed the wide variety of gas powered equipment available for air conditioning and described applications and advantages of each type.

PHILADELPHIA . . . Newly installed officers are: W. H. Mullin, President (July to December, 1960); Ludwig Mack, President (January to June, 1961); O. M. Kershock, 1st Vice President; D. S. Plewes, 2nd Vice President; A. A. Lincoln, Treasurer; and H. N. Teuber and P. R. Anderson, Recording and Correspondence Secretaries, respectively.

EVANSVILLE . . . Elected to office for the 1960-61 season were: R. E. Ahlf, President; H. C. Shagaloff, 1st Vice President; C. L. Herndon, 2nd Vice President; D. S. Phillips, 3rd Vice President; K. C. Davis, Treasurer; and D. Kuhlenschmidt, Secretary.

Underwriters' Laboratories was discussed by Norman H. Davis, Jr., of that organization, speaking at the May 3rd meeting. Stressing that performance of a device is not taken into consideration (except in the case of fire extinguishers), but only hazards in use, he showed a film, "Album of Public Safety," which briefly told the story of U. L.'s history and purpose. Scenes of tests on combustible materials, fireproof safes, building materials, electrical equipment, insulations, window air conditioners and refrigerator units were shown.

MONTREAL . . . D. M. Dymond of Canadian Standards Association was guest speaker at the May meeting, discussing CSA safety codes for fuel burners. Procedures, testing service and application of the codes to gas and oil burners were covered and composition and work of the organization were explained. Testing of a new burner to determine compliance with eleven code requirements was described in detail.

CENTRAL ARIZONA . . . Speaking at the May 2nd meeting on "Fan and Blower Rating by AMCA", Walter Curtis of Peerless Electric Div, H. K. Porter Company, outlined the goals of the organization as: standardization of sizes, dimensions and other such specifications; standard test codes; code of ethics for members; and certified rating program. Organization of the rating program was described, together with its functions and method of achieving enforcement of the program.

TOLEDO . . . Attempting to relate the apparent high cost of electric heating to the overall cost of owning and operating any type of heating system, R. L. Boyd made a strong case for electric heating at the May meeting. His discussion was accompanied by slides.

ST. LOUIS . . . Serving as Moderator of a technical forum on gas air conditioning, held at a recent meeting, was Charles Hartwein of Laclede Gas. Norbert K. Hall of Ready Power Company discussed

Among those in attendance at the Region X meeting in Fresno May 6th and 7th were: F. W. Hofmann, ASHRAE Assistant Secretary-Membership; J. M. McGrath and J. A. Hayes of California State Polytechnic Institute; J. P. Lamborn, President, San Joaquin Chapter; and R. H. Tull, ASHRAE Vice President.

Here gathered for the Region IX meeting, held May 4th in Salt Lake City, are: then-ASHRAE President Daniel D. Wile; Fred Janssen, Region IX Director; ASHRAE Vice President R. H. Tull; and F. W. Hofmann, ASHRAE Assistant Secretary-Membership.

"Gas Engine Driven Compression Systems", contrasting the economical merits of gas systems over electrically driven systems.

BALTIMORE . . . Pumps, their characteristics and selection, were under discussion at the May 5th meeting, when Tim Blong of Goulds Pumps, Inc., was guest speaker.

BOSTON . . . Speaker of the evening J. Gledhill of International Business Machines Corporation discussed cooling problems of computers at the May meeting.

SOUTHERN CALIFORNIA . . . Design and operation of the X-15 airplane environment control system was the subject under discussion at the May meeting, when Daniel G. Westbrook of North American Aviation was guest speaker. Stating that cooling system design temperatures were dictated by the electronic equipment in the plane, not by the needs of the pilot, he went on to explain that the latter wears a special suit and is able to withstand a much higher temperature than the equipment.

Used for cooling this plane is a cryogenic system utilizing liquid nitrogen as the refrigerant. Cabin and equipment required 71 lb of liquid nitrogen per flight, with a total of 135 lb being used for all of the plane's cooling needs, speaker Westbrook informed members. Describing the one-in. insulation blanket used in the plane, he stated that it has a "U" factor of 0.15 Btu/sq ft and has kept the inside surface temperature down to 200 F with an outer surface temperature of 1200 F during re-entry. Environmental test chambers were used extensively for testing components of the plane before any flights were made.

JOHNSTOWN . . . Moderating a panel discussion on specifications, held at the May 10th meeting, was W. W. Popyk of Crane Company. Members of the panel were: K. O. Schlentner of Crane Company, representing the manufacturer; H. F. Lenz of Lenz Engineering, representing the engineers; and C. V. Barnhart of Stanton-Barnhart, representing the contractor.

Stressed by Crane's Schlentner was the importance of having equipment in a specification meet standards set up by such organizations as SBI and AGA. Speaker Lenz covered briefly the legal aspects of a specification and contractor Barnhart concluded the discussion with a description of the type of specification that contractors prefer.

KANSAS CITY . . . Speaking at a joint meeting of this Chapter with the Home Builders Association and the American Institute of Electrical Engineers, Clark C. Anderson of Kansas City Power & Light Company discussed electric heating.

CENTRAL NEW YORK . . . Dominic D'Eustachio of Pittsburgh Corning Corporation reviewed the current ASHRAE research program and changes in its emphasis at the May 11th meeting. Following this, ASHRAE President Walter A. Grant discussed problems which have been encountered by National officers and committeemen as a result of the merger and general business conditions.

DALLAS . . . New officers for the 1960-61 season include: J. P. Jordan, President; R. M. Kilpatrick, 1st Vice President; R. A. Osterholm, 2nd Vice President; F. L. McFadden, 3rd Vice President; A. A. Hooper, Secretary; and O. Nation, Treasurer.

CINCINNATI . . . Covering the broad topic of air conditioning noise control, J. W. Overend of Koppers Company, Inc., guest speaker, divided his subject in two parts: "What are the pitfalls in determining noise control requirements?" and "How are noise reduction devices evaluated?"

MIDDLE TENNESSEE . . . "Reaction of Man to Inside Environments" was the topic presented by Burgess H. Jennings of the ASHRAE Research Laboratory, guest speaker at the May 10th meeting Reporting on current work of the laboratory on comfort conditions, he also covered effects of air conditioning as regards productivity and learning rates.

HOUSTON . . . Controls were covered in a panel discussion at the May meeting which took into consideration such aspects of the topic as applications, trends, sizing of valves and selection of various types of controls. Harold McKee of Powers Regulator Company moderated a panel comprised of William Flynn,

Minneapolis-Honeywell Regulator Company; Ken Alexander, Johnson Service Company; Donald Cole, R. H. Wittbold Company; and Walter Hainline, Powers Regulator Company.

ATLANTA . . . "Influence of Connections upon Fan Performance" was the subject of W. E. Tracy, Sturtevant Div, Westinghouse Electric Corporation, guest speaker at the April 11th meeting.

BATON ROUGE . . . Meeting jointly with local chapters of AIEE and ASME, members of this Chapter heard W. L. Budge of Westinghouse Electric Corporation's Atomic Power Dept speak on latest advances in nuclear energy.

MOBILE . . . Application of hospital air conditioning, with emphasis placed on the three-pipe induction system, was discussed by R. T. Lloyd of York Corporation at the May 23rd meeting.

ROCHESTER . . . Don Lane of American-Standard discussed application of fans to various industrial problems and John Lents of Kenmore Machine Products spoke on use of silica gel and molecular sieves for moisture control in refrigeration systems at the April 6th meeting.

COLUMBUS . . . Highlighting the April 18th meeting was a plant trip through Western Electric Company.

CENTRAL ARIZONA . . . Speaker at a recent meeting was R. H. Phillips of Carrier Corporation, who presented a discussion of absorption refrigeration for air conditioning.

NEW ORLEANS . . . Members attending the May 17th meeting heard J. J. Landry, Supervising Engineer of Southern Bell Telephone & Telegraph Company, discuss "Developments in electronics affecting communications".

NORTHEASTERN OKLAHOMA . . . Elected to office for the 1960-61 season were: John Nichols, President; James Shipman, Vice President; Virgil Carrier, Secretary; Warren Smith, Treasurer; and Clarence Dollmeyer, William Buckner and Jack Tumilty, Board of Governors.

MISSISSIPPI . . . New officers for the 1960-61 season include: William D. Fortner, President; Charles E. Strahan, Jr., Vice President; H. Stuart Thomas, Secretary; H. Tom Allen, Treasurer; and Forrest H. North, Frank King and Carlan H. Schmalz, Board of Governors.

Constantin P. Yaglou

1896-1960

Long-time Professor of Industrial Hygiene at Harvard University School of Public Health, Constantin P. Yaglou died June 3, 1960. He was the 1959 recipient of the F. Paul Anderson Medal, given in recognition of outstanding work in his field. Long engaged in teaching and research on hygienic aspects of air conditioning, he had contributed many papers on various phases of this subject for publication in engineering, medical and public health journals.

Born in Constantinople, Turkey, on January 26, 1896, he attended the French-Greek College of Arts and Sciences, later graduating from the Engineering School of Robert College with a degree of B. S. M. E. His formal education was completed at Cornell University, where he received his M. M. E. degree.

From 1921-25 he served on the staff of the former ASHVE Research Laboratory. During these years he was associated closely with most

of the pioneering studies in development of the original effective temperature scales, determination of the comfort zone and physiological effect of heat and cold. With the School of Public Health since 1925, he had conducted further studies in the comfort, health and therapeutic aspects of air conditioning and on working efficiencies in climatic extremes.

A member of the former ASHAE and precedent ASHVE since 1923, Professor Yaglou served on the Guide Publication Committee in 1926, 1946 and 1947, and on the Nominating Committee as a delegate from Massachusetts Chapter in 1932 and 1938. Active on numerous Research Technical Advisory Committees, he was Chairman of the committees on Comfort Air Conditioning, Re-study of Comfort Chart and Comfort Line and Sensations of Comfort and a member of Comfort Standards for Summer Cooling, Instruments, Navy Studies, Human Calorimetry and Physiological Research.

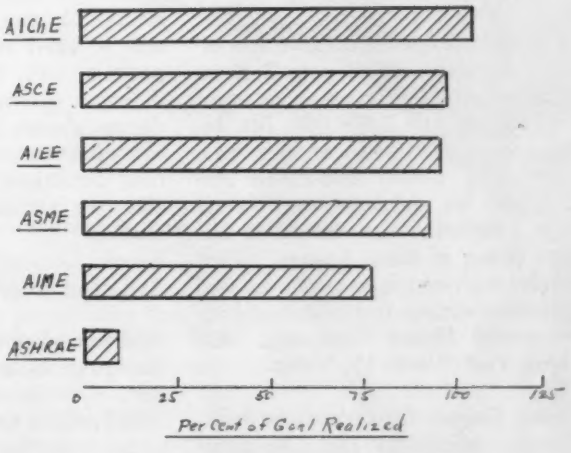
Local Chapter activities included serving on the Board of Governors of Massachusetts Chapter in 1936 and 1941, as Vice President in 1937 and 1939 and President in 1940.

UEC Cornerstone Is Set

On June 16, before distinguished guests and representatives of those engineering societies which will occupy the United Engineering Center upon its planned completion next year, the cornerstone of UEC was set with appropriate ceremonies. Mayor Wagner of New York and U. S. Former President Herbert Hoover were scheduled participants.

Within the stone was sealed a copper box containing memorabilia of the constituent societies. ASHRAE material consisted of a recent copy of the JOURNAL, a printing of the By-laws and an informative folder.

The campaign for funds continues to be pushed actively. Latest figures for ASHRAE show $23,222 from 577 contributors, against a goal of $250,000. The member fund total is $3,145,500 against a $3,800,000 objective.



As indicated in the accompanying bar chart of percentage of quota accomplishments, our record does not yet reflect appropriate results for efforts already expended.

For your convenience a contribution pledge form is incorporated within this page. Mail to ASHRAE, 62 Worth Street, New York 13, N. Y.

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BULLETINS and CATALOGS

Shaded Pole Fan Motors. Four-page Bulletin GEC-219F discusses applications and features of 51-frame, unit-bearing, totally enclosed fan motors. Advanced lubrication, bearing and insulation systems on the 3½-in. diam motor are described and testing procedures, construction, delivery, service and ordering directions are outlined. Tables on motor ratings, sizes and fan blades, as well as motor dimensions and operating data, are included.
General Electric Company, Schenectady 5, N. Y.

Noise Control Specifications. Designated Bulletin F2D, this six-page guide to writing vibration and noise control specifications for air conditioning and related equipment discusses factors involved in selection of the various resilient mountings commercially available, with particular attention to effects of existing job conditions, specifically extremely critical, critical and non-critical conditions. Illustrations show the type of vibration control mounting to use for each problem.
Korfund Company, Inc., 48-53D 32nd Pl., Long Island City 1, N. Y.

Hot Water Heaters. Extensive specifications are given in four-page Bulletin 77 for nine models ranging in input from 180,000 to 1,800,000 Btu/hr. Cut-away drawings show fire tube construction, quality controlled system, water leg and integrated burner design. Bulletin also describes the nickel lining of these heaters, which provides high corrosion resistance and non-porous surface to prohibit scaling.
Commercial Heater Company, 3020 Galvez, Fort Worth 11, Texas.

Cooling Towers. Blow-through, draw-through, centrifugal fan and tube-axial fan models from ten to 700 ton in single, factory-assembled units are discussed in 28-page Bulletin 207. Presented are engineering drawings, capacity and performance data, selection tables and product descriptions.
Baltimore Aircoil Company, Inc., P.O. Box 7322, Baltimore 27, Md.

Dampers. Drafstop, a new design back draft damper having a free area of 75%, uses flexible blades of either synthetic rubber-coated nylon or Teflon-coated glass fabric. Available in all sizes for either high or low velocity applications in ventilating and air conditioning systems, it is the subject of four-page Bulletin 105.
Airflow Controls Company, 621 W. Lake St., Minneapolis 8, Minn.

Blast Coil Space Heaters. Edition nine of Bulletin E-97U contains 32 pages of information on this open coil duct heater, which is cited as being easily installed; requiring no floor space, wall space or flue; and having been proven maintenance-free. Stock sizes are shown from 1000 to 30,000 watt (rating of 3400 to 102,000 Btu/hr) and custom sizes from 500 to 1,000,000 watt (rating of 1700 to 3,412,000 Btu/hr).
Industrial Engineering & Equipment Company, 24 Hanley Industrial Ct., St. Louis 17, Mo.

Coil Selection Tables. Performance of all units in the air conditioning and Multizone lines is shown in this set of coil selection tables, which may also be used as a reference for applications in water cooling, direct expansion and heating with water and steam. Cooling performance for each unit is listed at five different air capacities, with further data for three different entering chilled water conditions shown for each air capacity, as well as three evaporating temperature conditions for direct expansion and two air entering conditions for heating.
Recold Corporation, 7250 E. Slauson Ave., Los Angeles 22, Calif.

Resilient-Mounted Motors. Listed in four-page Bulletin MU-245 are motors, 1/6 through ten hp, available with resilient mountings. Booklet discusses mounting construction and various problems involved in quiet motor operation.
Wagner Electric Corporation, 6400 Plymouth Ave., St. Louis 33, Mo.

Home Heating and Cooling. Illustrating this manufacturer's line of water heaters, coolers, pumps, valves, control units and other component products for heating and cooling homes, 25-page Catalog GC-1259A diagrams key models in each of the product categories. Accompanying technical data detail functions performed, performance capacity range, materials of construction and special features offered. Described in the catalog are water heating units, booster pumps, compression tanks, air control units, air vents, tank and tankless water heaters, air chargers, heat exchangers and water mixers.
Bell & Gossett Company, 8200 N. Austin, Morton Grove, Ill.

Pressure Regulators. Giving currently available information on sliding gate pressure regulators, eight-page Catalog J160-1 describes in detail self-operated, pilot-operated, solenoid-operated and back pressure regulators. Applications of each of the four basic types and how each works is presented and the self-cleaning, self-lapping action of the sliding gate seat is illustrated. Features, cut-away drawings, sizing charts, flow curve and sample specifications are included.
OPW-Jordan Corporation, 6013 Wiehe Rd., Cincinnati 13, Ohio.

Refrigerant Condensing Method. Equipment for uniformly controlling condensing temperatures and pressures under all dry bulb conditions is introduced in six-page Bulletin DCR-60. System consists of a basic air-cooled condensing unit, pressure controlled damper system for low dry bulb operation and an air precooler that lowers high dry bulb entering temperature.
Marley Company, 222 W. Gregory Blvd., Kansas City 14, Mo.

Circuit Breaker Test Units. Low voltage power circuit breakers and molded-case circuits can readily be tested with these instruments. Features and advantages of several integrated units are presented in 16-page Bulletin SB-CB-1. Contained are typical test procedures, general considerations for testing, selector chart of test unit models for various sizes of circuit breakers, descriptions and specifications, method of operation and a list of possible applications.
Multi-Amp Electronic Corporation, P. O. Box 217, Union, N. J.

Fluid Drives. Descriptive of the Class 4 line of adjustable speed Gyrol fluid drives is 20-page Bulletin A-719. Discussed are principles of operation, typical applications and advantages of adjustable speed fluid drives, with hp and inertia starting capacity curves used for graphic presentation. Components of the unit are illustrated and described and direct-reading selection guide charts are presented for both variable and constant torque

(*Continued on page* 78)

Using a candle to test the quality of eggs is one thing . . . but to detect flaws in thin-walled copper tubing, Viking relies on the latest electronic equipment and scientific principles.

For example, every inch of straight-length tubing at Viking passes through a sensitive electronic "brain" built to Viking specifications. This machine detects the tiniest pinhole, the minutest crack, bump or out-of-round. If a defect is present the "brain" automatically marks the tubing and rejects it.

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VIKING

COPPER TUBE CO.

CLEVELAND 10, OHIO

A COMPARATIVE STUDY OF MANUFACTURING COSTS

(*Continued from page 41*)

Fixed cost of cooling unit = F
Fixed cost of power supply = f

$$\text{Current dependent power supply cost} = \sqrt{\frac{Q\,Ea\,\beta\,T_c\,\alpha}{2K\rho\,(2\beta\,T_c^2Z - \beta^2\,T_c^2Z - 2\Delta T)}}$$

$$\text{Power dependent power supply cost} = \frac{Qb\,(2\beta^2\,T_c^2Z + 2\beta\,T_c\,\Delta T\,Z)}{(2\beta\,T_c^2Z - \beta^2\,T_c^2Z - 2\Delta T)}$$

$$\text{Assembly cost} = \sqrt{\frac{Q\,Ea\,\beta\,T_c\,\alpha}{2K\rho\,(2\beta\,T_c^2Z - \beta^2\,T_c^2Z - 2\Delta T)}}$$

$$\text{Semiconductor material cost} = \frac{2\,Q\,L^2M}{K\,(2\beta\,T_c^2Z - \beta^2\,T_c^2Z - 2\Delta T)}$$

The final expression shows that the material cost is proportional to L^2 as previously mentioned.

These equations are now applied to specific capacities and the results are shown in Tables III, IV, V and VI and Fig. 3.

Some general observations can be drawn from Fig. 3. They are as follows:

1. For large values of β the cost changes slowly, whereas, for small values of β the cost changes rapidly.

2. The range over which the cost remains fairly constant decreases as:
   a. The operating temperature difference increases.
   b. The design capacity of the device increases.

3. Considerably enhanced C.O.P. can be achieved with little increase in cost by operating couples at somewhat less than their maximum heat pumping capacity.

4. The design to achieve minimum costs becomes more critical at higher capacities. For example, if one can extrapolate to a 10,000 Btu per hr capacity a change in β from 1.00 to 0.8 represents a saving of $48.00 or 4.4%.

5. The arbitrary choice of A/L = 0.5 Cm selected for the preliminary analyses is probably close to optimum design.

6. By consideration of the contribution to the total cost of a system by the basic material used and the power supply it can readily be seen where cost reduction must be sought. For example, by referring to the minimum cost figures we see that for the 1,000 Btu per hr unit the power supply and TE material comprise 86.7% of the total cost (26.7% and 60% respectively). For the 100 Btu per hr unit the corresponding figures are 67% (36% and 31%). By merely reducing the quantity of material employed in a device by a factor of 2 (L reduced by $\sqrt{2}$) the manufacturing cost curve for $Z = 3.0 \times 10^{-3} C^1$ becomes almost identical with the curve shown for $Z = 5 \times 10^{-3} C^1$. This is indicative of the possibilities for future cost reduction.

Contact resistance has been considered to be negligible in this

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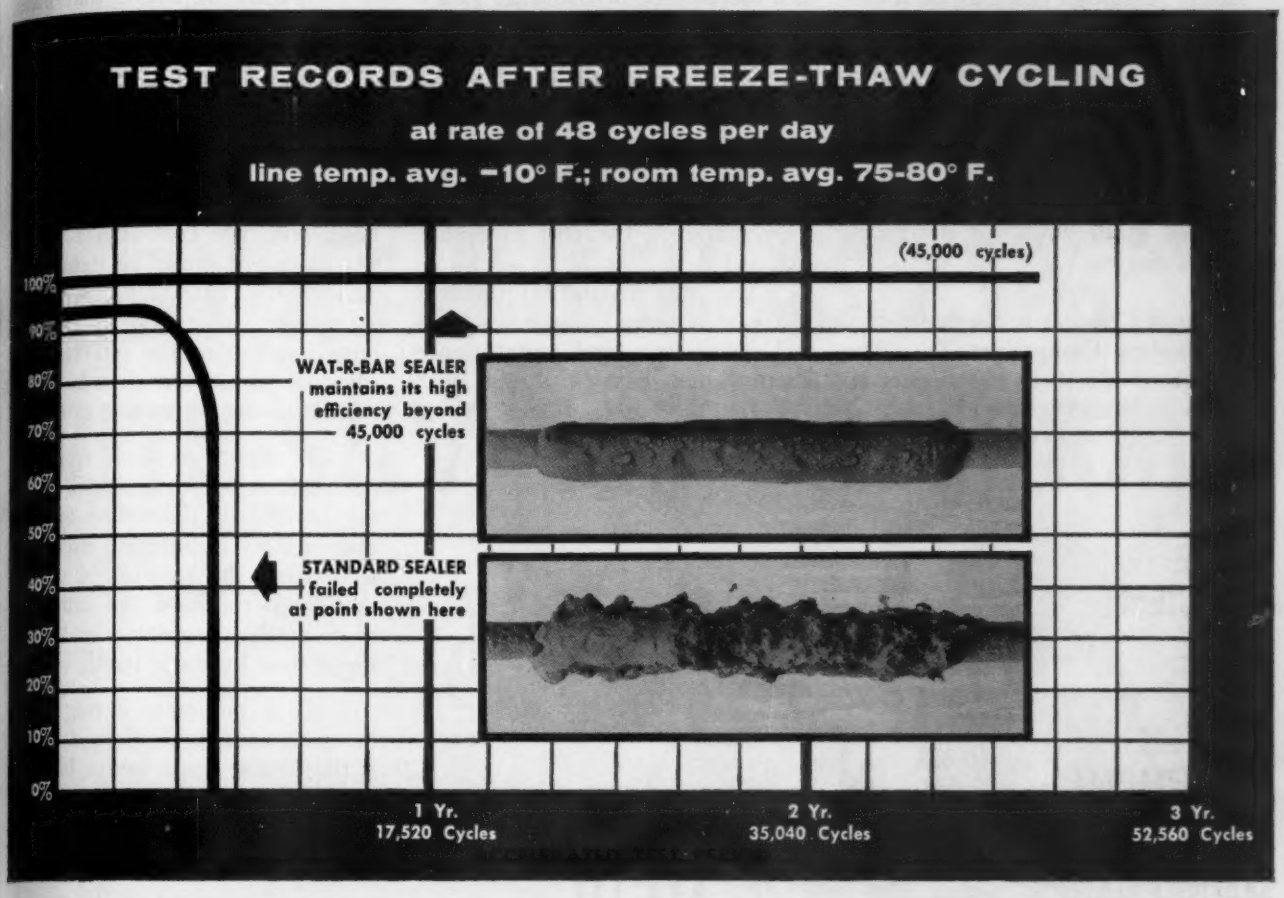

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study. Although this may be somewhat optimistic its omission does not seriously affect the general conclusions. The method employed here is a general one applicable for any cost figures and any capacity device operating at any desired condition. It consequently provides helpful guide lines for design of future devices.

Effect of Changes in the Individual Parameters Comprising Z – The Figure of Merit, Z, has been considered the all important parameter of a thermoelectric material. This is true from considerations of performance since it determines the coefficient of performance under any operating conditions. However, when one re-examines the cost equations the individual parameters α, ρ and K become important, since they influence the cost independently of Z.

The 3rd, 5th and 6th terms in the cost equation contain individual parameters and consequently their values have a direct bearing on the cost of the device. It is readily apparent that materials having high values for α and K are desirable. For example, if α changed from 400 to 800 μV/C and K from 1.6×10^{-2} to 6.4×10^{-2} $Wcm^{-1} C^{-1}$, with Z remaining at $3 \times 10^{-3} C^{-1}$ in going from one material to another having the same basic cost, the cost of a 1000 Btu per hr system could be reduced by almost 50%. Although this may be a grossly exaggerated example, consideration of the effects of individual parameters does bring out the following important points:

(1) Although Z is important in determining performance it does not necessarily determine the total cost of the TE materials used. This is because the amount of the TE materials required is influenced more by the parameters making up the Z than by the Z itself.

(2) A moderate Z might well be more acceptable than a high Z if the former has a better blend of parameters.

(3) Work on compounds made from expensive elements should not be discouraged on the grounds of cost, since the possibility exists of such a material not only having an improved Z, but also of having a blend of parameters which would reduce the cost of a device in such a way as to more than offset the increase in basic material cost.

(4) If sufficiently high Z values are obtained in the future, then the opportunity of doping such a material in a way such as not to realize its full potential but to obtain more favorable individual parameters from the cost viewpoint might present itself.

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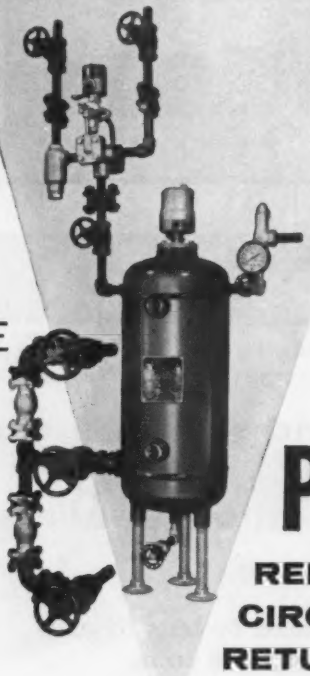

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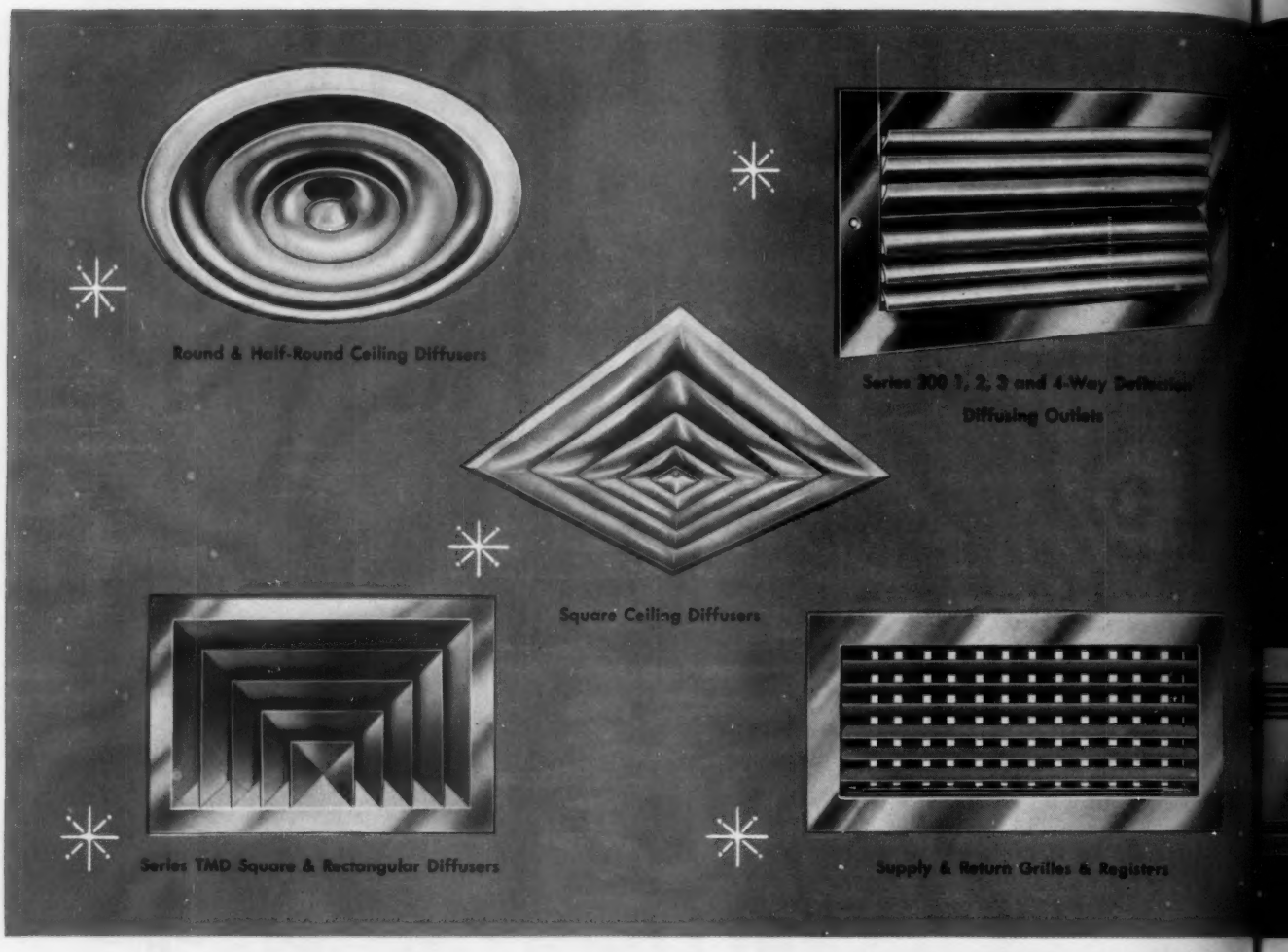
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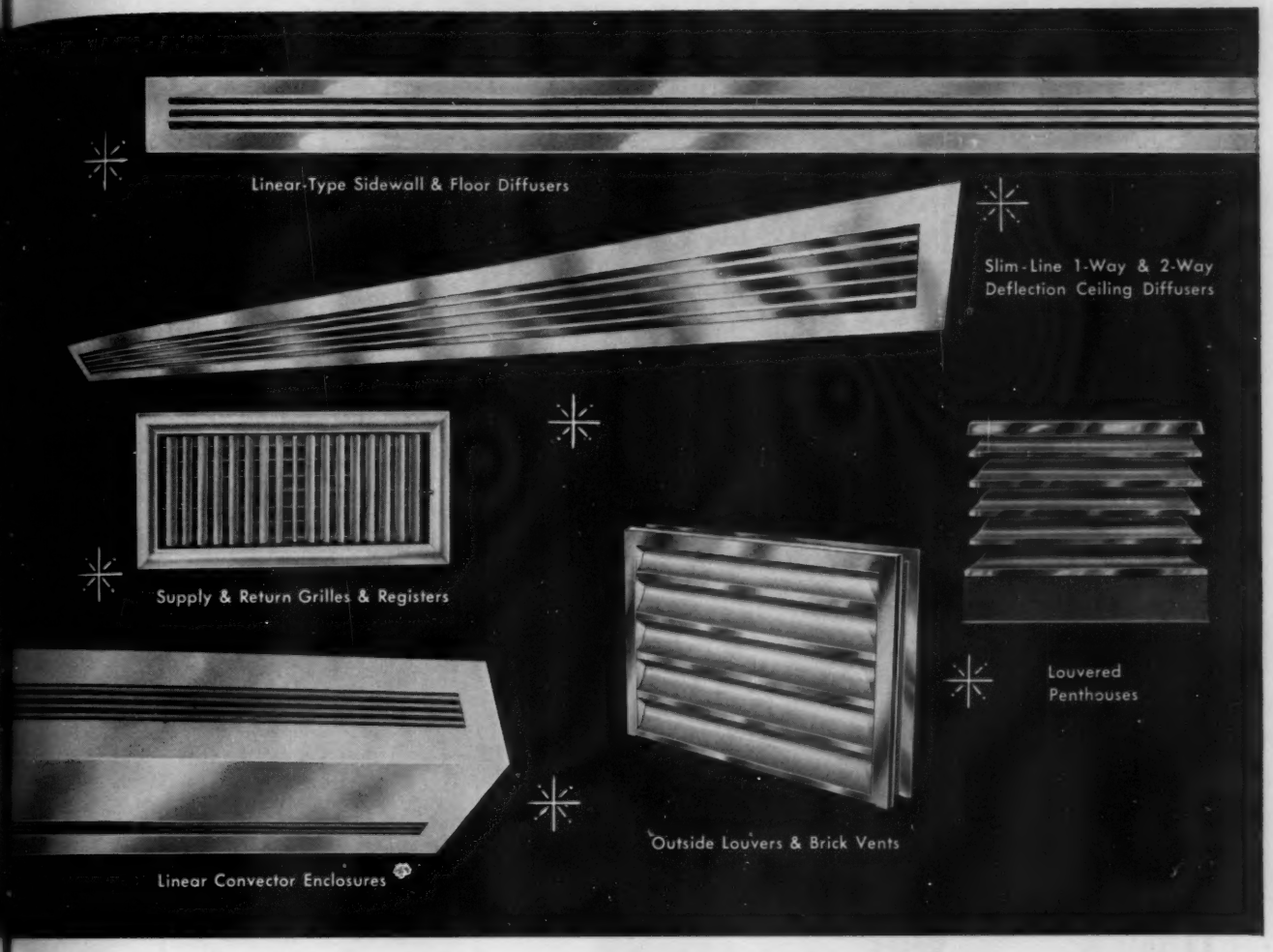
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BULLETINS

(*Continued from page 70*)

applications. Dimensional data is provided in tabular form and correlated with layout type dimensional drawings.
American Radiator & Standard Sanitary Corporation, Industrial Div, Detroit 32, Mich.

Evaporative Condensers. Covered in 24-page Bulletin 107 is an extensive line of evaporative condensers in capacities from ten to 700 ton in single, factory-assembled units. Engineering drawings, capacity and performance data, selection tables and product descriptions are given for blow-through, draw-through, centrifugal fan and tubeaxial fan models.
Baltimore Aircoil Company, Inc., P. O. Box 7322, Baltimore 27, Md.

Variable-Speed Sheave. Revised to provide additional technical data and new list prices on the SVS variable-speed sheave is eight-page Bulletin 6102. Sheave is adjusted to desired pitch diam by turning an adjusting screw. Continuous speed control is cited as being provided without sticking or freezing.
T. B. Wood's Sons Company, Chambersburg, Pa.

Product Cooler. Information on vertical and horizontal product coolers with centrifugal blowers is contained in Catalog 8COa. These units feature modular construction and allow for a great flexibility in location of blower discharges, motor and coil. Coolers are available for either high temperature (above 34 F) or below freezing applications, with either water or gas defrost. Offered in the catalog is information on dimensions, arrangements and performance.
Recold Corporation, 7250 E. Slauson Ave., Los Angeles 22, Calif.

Motor Application Guide. Outlining motor characteristics and including recent information on motor designs is 16-page Bulletin 270B. Motors are illustrated and information on torques, load characteristics, starting current, speed, capacity, performance and mounting is presented.
Century Electric Company, 1806 Pine St., St. Louis 3, Mo.

Refrigeration Products. Catalog R460, in addition to listing an extensive line of flare tube fittings and such items as hermetic port valves, color coded charging lines, driers, strainers and capillary tubing, also shows this manufacturer's recently introduced products, such as "tamper proof" tube piercing valves, small general purpose valves, testing outfits and thread sealer.
Madden Brass Products Company, Aurora, Ill.

Circuit Breaker Load Centers. Descriptive information and specifications on 15 new 150 and 250-amp circuit breaker load centers are provided in 16-page Bulletin GEA-7071. Also featured is a load center line extending from 2 to 40 circuits, 40 to 200 amp in flush, surface and raintight enclosures.
General Electric Company, Plainville, Conn.

Air Conditioners. Provided in six-page Catalog 310 are breakdown descriptions of each of the four sizes of Luxaire Air-Cooled Condensing Units, two, three, four and five-hp. Also illustrated and described are round or oval, plenum-type, duct-type and counterflow cooling coils. Two pages of diagrams and tables show dimensions, ratings and specifications of these units.
C. A. Olsen Manufacturing Company, Elyria, Ohio.

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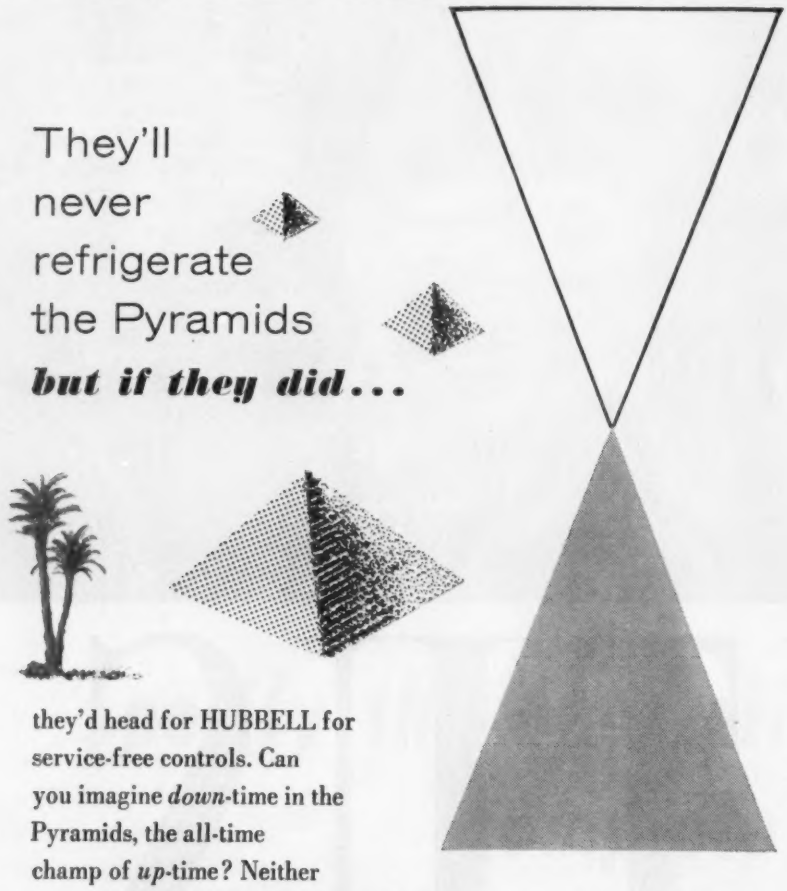
but if they did...



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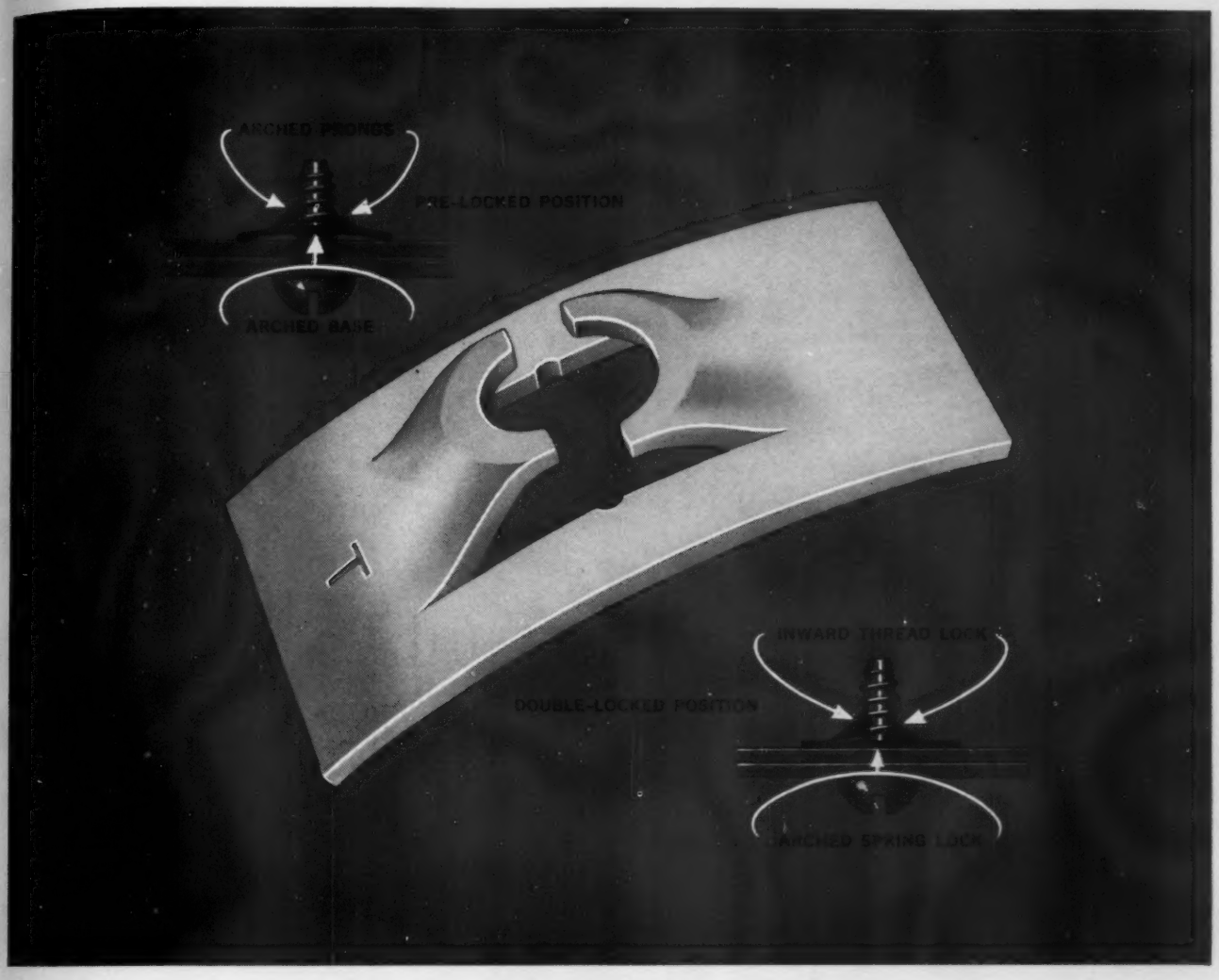

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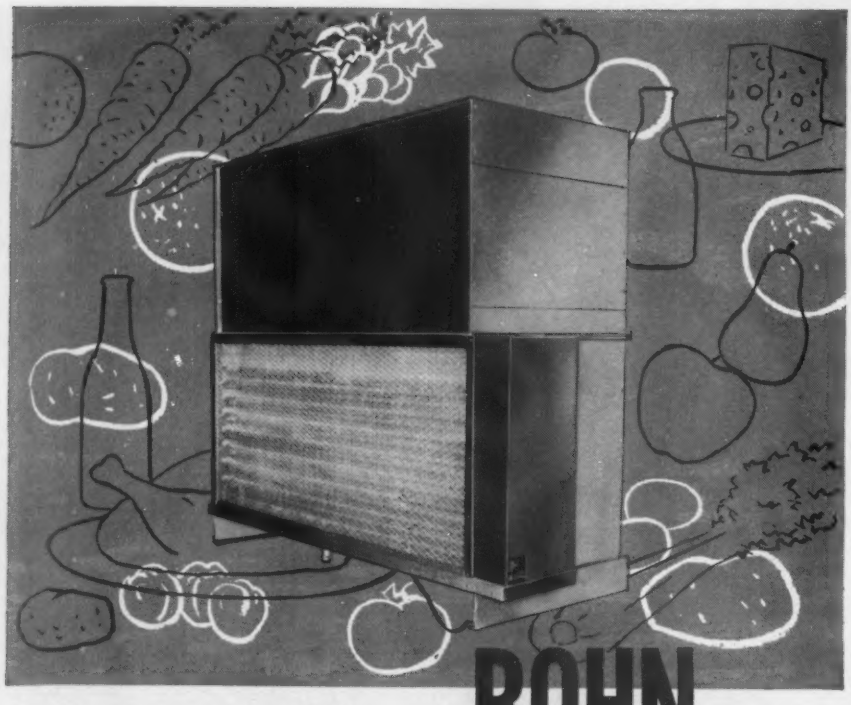

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CAPACITY DATA

Model Number		BTU/Hr.	CFM	Shp. Wgt.
*Vertical	Horizontal	10° T. D.		Approx.
5800PCV-1	5800PCH-1	58,000	8,300	1,000 lbs.
7500PCV-1	7500PCH-1	75,000	10,600	1,200 lbs.
9200PCV-1	9200PCH-1	92,000	13,100	1,400 lbs.

*Discharge ducts available for vertical models.

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Others are saying

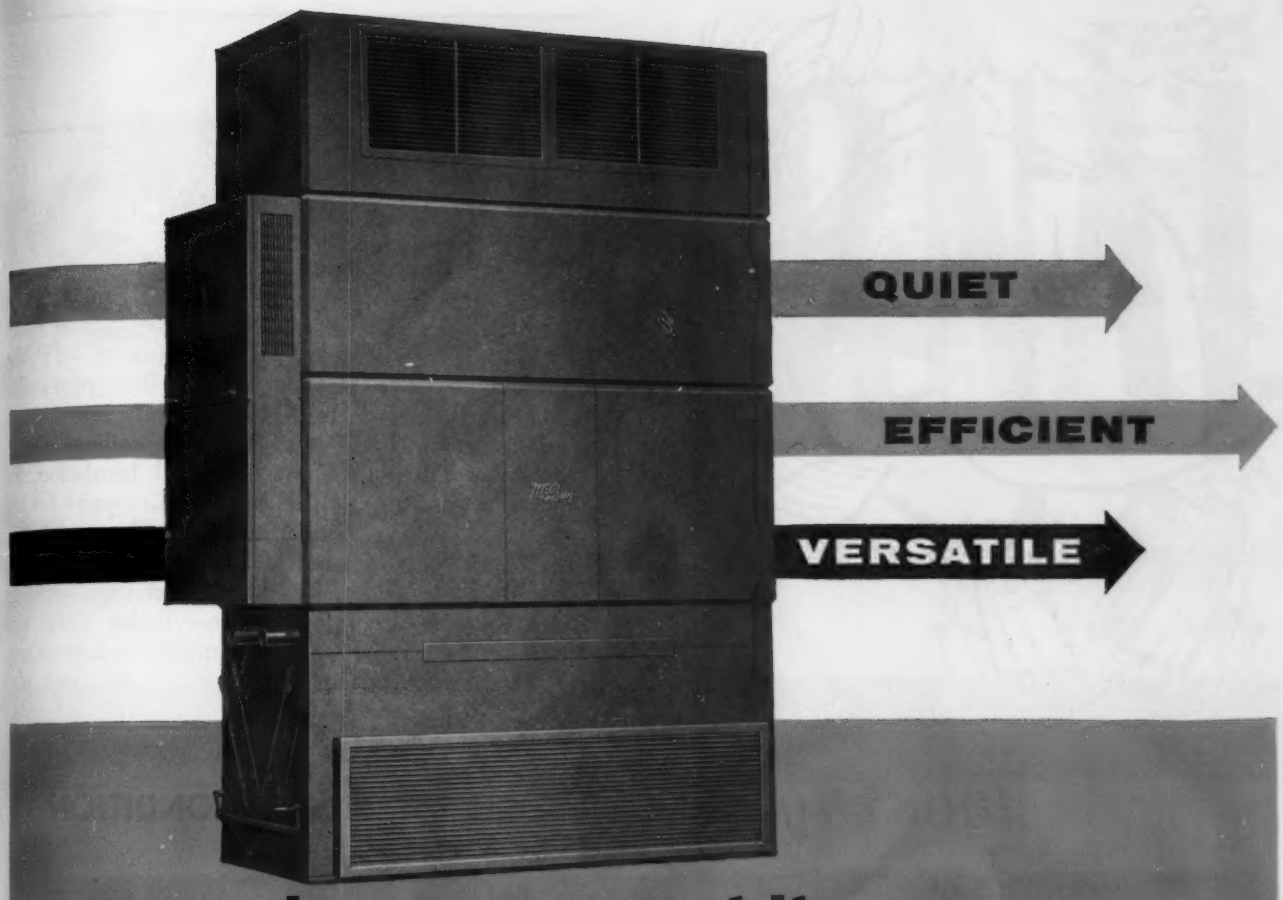
(*Continued from page 61*)

windows were closed. *Journal of the Institution of Heating and Ventilating Engineers, May 1960, p 57 (British).*

that . . . . in applications such as paint spray shops where it is necessary to maintain a high rate of air extraction from an enclosed space in order to counteract air contamination, heat loss is usually quite considerable and will lower the ambient temperature not only of the shop where the extraction fans are located, but also of any adjoining premises from which replacement air may flow. Normal solution to this problem is installation of additional fans and heaters capable of restoring a volume of fresh air at shop temperature. Heating make-up air directly from oil without resorting to use of boiling water or steam is cited as offering numerous advantages. *Heating, April 1960, p 111 (British).*

that . . . . developed to improve the sterility of air in the vicinity of an operating table, this new approach to distribution of conditioned air bathes the patient in a current of fresh supply air, providing a protective curtain about the operating field. Ideal air distribution would be one in which the entire ceiling is an air supply and the entire floor an exhaust. Because of the impracticability of this, the operating facilities at this cited hospital will have a modified version in which the air supply is directly over the operating table and an exhaust opening is provided near the floor on all four walls. *Architectural Record, April 1960, p 251.*

that . . . . . results of air pollution testing made by the Public Health Service's National Air Sampling Network indicate a geometric mean particulate concentration over urban areas of 136 microgram/cu meter as compared with 33 for non-urban areas. In terms of settlable dust, sampling over a period of years has shown the effect of control measures, but the curves have levelled off, indicating limits imposed by practicability and economics of further removal. *Public Health Reports, May 1960, p 413.*



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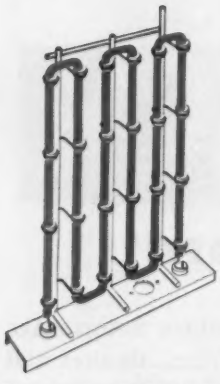

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Applications

NEW INSULATION METHOD USED IN WAREHOUSE

In remodeling cold storage facilities at the Point Adams Packing Company, Newport, Ore., to maintain a constant warehouse temperature of –8 F, the contractor selected a new water-repellent vermiculite insulation that could be poured directly into existing wall cavities and the area above the ceiling. Designated Water-Repellent Masonry Fill Insulation, the product was developed by Zonolite Company for use in cement block or brick cavity walls.

Refrigeration system for the warehouse is high pressure with an ammonia base. Three four-cylinder 15-hp compressors and one two-cylinder 25-hp unit comprise the installation. Refrigerant conductors consist of direct expansion coils of 1½-in. steel pipe located in the ceiling.

RADIANT CEILINGS CONDITION LONDON BUILDINGS

Installed in the Shell Center, a related group of office buildings in London, England, approximately 1,000,000 sq ft of radiant acoustical ceiling provide heat, sound absorption and summertime cooling to offices. Manufactured by Burgess-Manning, the ceiling operates by circulation of water through sinuous coils supporting perforated aluminum panels which radiate the heat. Chilled water is pumped through the coils in summer and thermostats in each office provide flexibility of control.

MOISTURE CONTROL AIDS BEARING PRODUCTION

Contamination and excessive humidity in the main plant of Fafnir Bearing Company, New Britain, Conn., caused abrasion and microscopic rust on polished bearing surfaces. To solve this problem, all air introduced into assembly and inspection areas through the air conditioning system is filtered both electrostatically and by regular cleanable filters. Also provided is a high rate of air exchange to remove contaminated air as soon as possible. To prevent formation of bearing rust, the air conditioning system is designed to hold 45% relative humidity in the assembly area and 40% relative humidity in the inspection area.

Air handling equipment consists of two Trane Climate Changer central station air conditioners, one to serve the assembly area, the other handling the inspection area. Eight pre-cooling and reheat coils are incorporated into the system for better zone control. Chilled water for the cooling coils is provided by a 300 hp Trane Hermetic CenTraVac centrifugal compressor.